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THE BABCOCK AND WILCOX HIGH PRESSURE SECTIONAL EXPRESS BOILER.

BY LIEUTENANT-COMMANDER R. C. BRIERLY, U. S. N. R., MEMBER.*

NOTE: An article by Mr. Brierly entitled "Four Boilers Manufactured by the Babcock and Wilcox Company Tested at the Fuel Oil Testing Plant" appeared in the November, 1929, issue of the JOURNAL. At that writing only a small portion of the test operation on this boiler had been completed. The present article discusses results and observations from the test which was concluded early this year.

THE BOILER.

The High Pressure Sectional Express boiler used during the test herein discussed was designed by the Babcock and Wilcox Co., 85 Liberty Street, New York City, New York. The parts were manufactured at their works at Barberton, Ohio, and assem-

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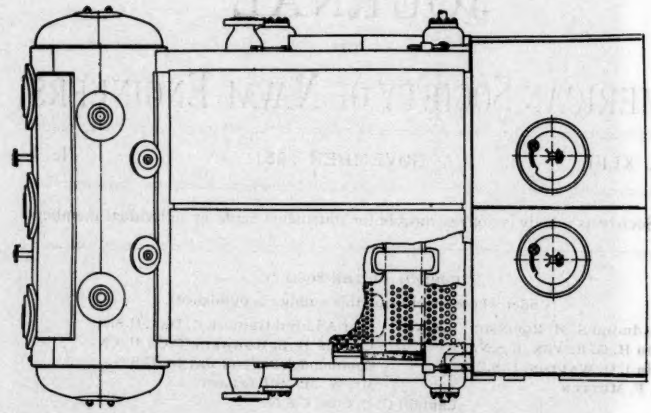


FIGURE 2.

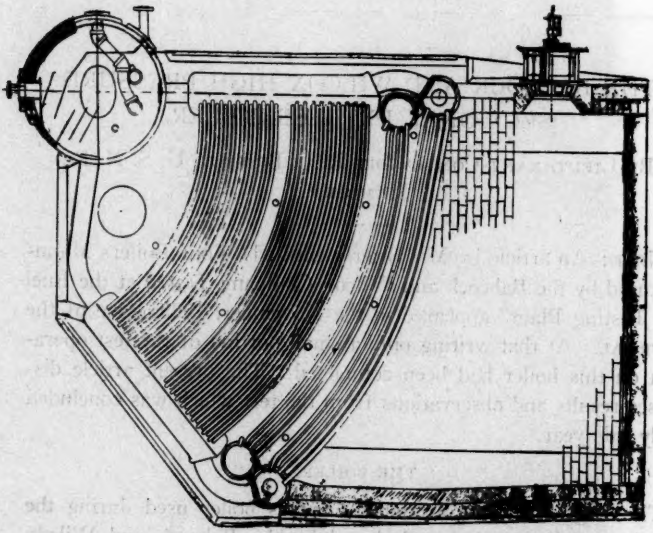
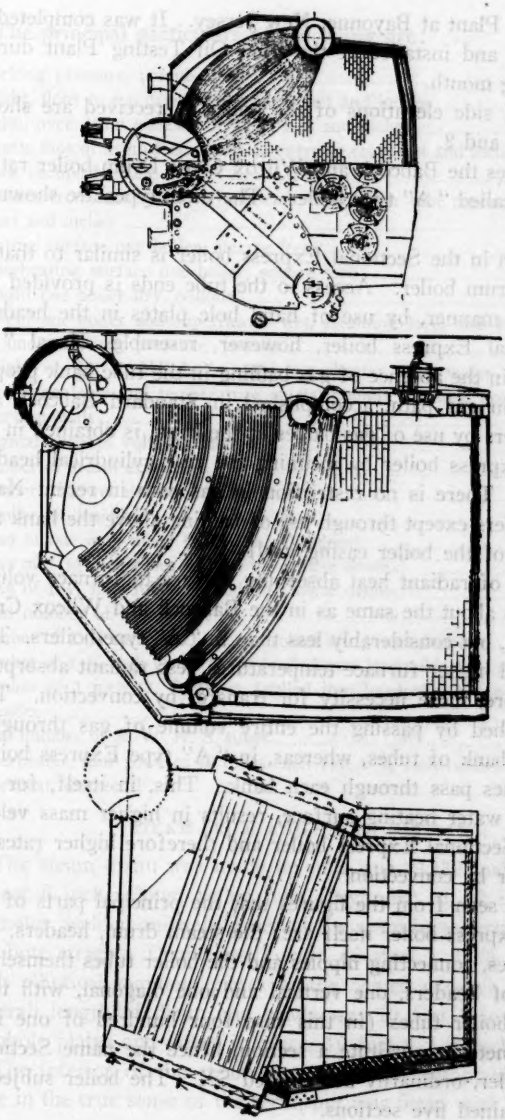


FIGURE 1.



BABCOCK and WILCOX CROSS DRUM BOILER

BABCOCK and WILCOX SECTIONAL EXPRESS BOILER

BABCOCK and WILCOX "A" TYPE EXPRESS BOILER

FIGURE 3.

bled at their Plant at Bayonne, New Jersey. It was completed in April, 1929, and installed at the Fuel Oil Testing Plant during the following month.

Front and side elevations of the boiler as received are shown in Figures 1 and 2.

It resembles the Babcock and Wilcox Cross Drum boiler rather than the so-called " A" type boiler. The three types are shown in Figure 3.

Circulation in the Sectional Express boiler is similar to that in the Cross Drum boiler. Access to the tube ends is provided for in the same manner, by use of hand hole plates in the headers. The Sectional Express boiler, however, resembles Naval " A" type boilers in the absence of gas baffling in the tube bank proper. Restriction in gas path, accomplished in Standard Babcock and Wilcox boilers by use of gas baffles in the bank, is obtained in the Sectional Express boiler by inclining the rear cylindrical headers and casing. There is no restriction in gas path in recent Naval Express boilers except through use of baffling above the bank and the contour of the boiler casing itself.

The ratio of radiant heat absorbing surface to furnace volume in the SX is about the same as in the Babcock and Wilcox Cross Drum boiler, but considerably less than in " A" type boilers. This tends toward higher furnace temperatures, less radiant absorption and, therefore, more necessity for transfer by convection. This is accomplished by passing the entire volume of gas through a single deep bank of tubes, whereas, in " A" type Express boilers half the gases pass through each bank. This, in itself, for the same boiler water heating surface, results in higher mass velocities in the Sectional Express boiler and therefore higher rates of heat transfer by convection.

It will be seen from the figures that the principal parts of the Sectional Express boiler itself are: the steam drum, headers, cylindrical boxes, connecting nipples and the boiler tubes themselves. Each pair of headers, one vertical and one diagonal, with their connecting boiler tubes (in this case four hundred of one inch outside diameter) constitute a section, hence the name Sectional Express boiler, ordinarily abbreviated SX. The boiler subjected to test contained five sections.

The principal particulars of the boiler are:

Working pressure, pounds	600
Height, floor to top drum over rivets, feet and inches	17 0
Width, over drum manhole fittings, feet and inches	10 7
Length, face of drum feed nozzle to extreme rear, feet and inches	13 6¼
Space occupied, cubic feet	2433
Distance from front plate to bulkhead required for tube renewal, feet and inches	10 8
Heating surface one boiler, square feet	5083
Superheating surface one boiler, square feet	516
Weight one boiler dry, pounds	82,250
Weight one boiler and superheater wet, steaming level and 600 pounds working pressure, pounds	89,850
Weight of the SAME boiler plus superheater with water at steaming level and temperature at 400 pounds working pressure, pounds	90,150
Original furnace volume, cubic feet	423.6
Number of steam drums	1
Number of lower drums	2
Number of headers	10
Rows of one inch tubes (rolled into headers)	38
Rows of 1 ¼ inch tubes (rolled into lower drums)	3
Rows of 1 ½ inch tubes (rolled into lower drums)	2
Total number of tube rows	43
Number of 1 ½ inch O. D. tubes (rolled into lower drums)	70
Number of 1 ¼ inch O. D. tubes (rolled into lower drums)	125
Number of 1 inch O. D. tubes (rolled into headers), 400 per header	2000
Total number of B. W. H. S. tubes	2195
Superheater, four rows of 40 tubes each (all 1 ¼ inch tubes)	160
Superheat, degrees F	200–300

STEAM DRUM AND STEAM BAFFLING.

The steam drum was made up of a one 21/32 inch steel plate, 8 feet 6 inches long, rolled into a cylinder of 42 inches inside diameter, welded and reinforced at the seam by external and internal butt straps. Drum heads were of steel 1 3/16 inches thick with elliptical manhole plates at the center of each drum head. Overall length of drum and heads was 10 feet 1 inch, and with manhole plates and yokes in place, 10 feet 4 1/2 inches.

The interior of the steam drum was not equipped with a dry pipe in the true sense of the term, but was fitted with a perforated

plate passing practically the entire drum length below the main stop and safety valve openings.

The steam drum also was originally equipped with a main steam baffle constructed of a series of plates extending practically the entire length of the drum. Flanges were provided at the ends of this baffle and end plates were bolted in place before runs on the boiler were begun. Three plates or shutters in the wake of — and above — the circulating tube exits had been installed prior to receipt of the boiler. An additional "dry plate" was fitted below the perforated steam exit baffle to prevent direct flow of steam through the perforations. Arrangement of this baffling as originally installed is shown in Figure 4.

ARRANGEMENT OF STEAM BAFFLES AS ORIGINALLY INSTALLED.

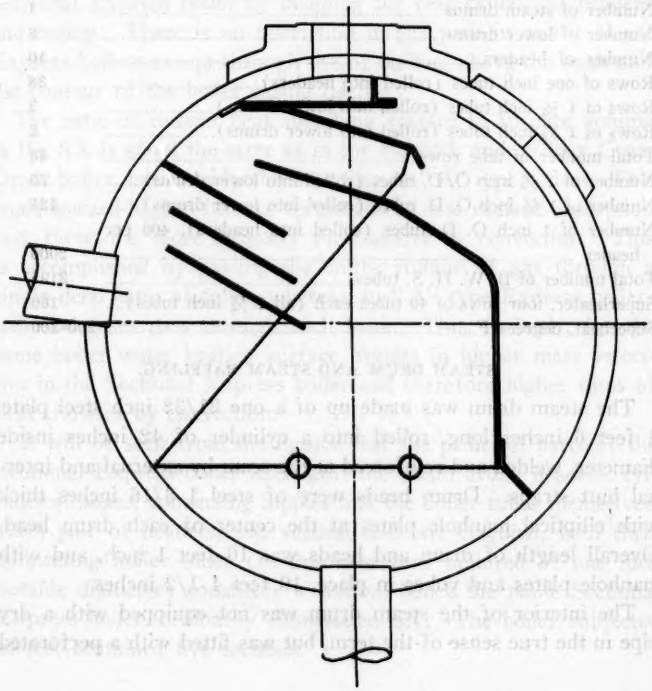


FIGURE 4.

SUPERHEATER.

The superheater, consisting of four rows of 1 1/4 inch tubes rolled into two cylindrical superheater boxes, was located in the space between the lower ends of the sections and the tubes connecting the cylindrical boxes below them. There were five rows of tubes between the superheater and the furnace. The "A" and "B" rows were 1 1/2 inches O.D.; "C," "D," and "E" were 1 1/4 inches O.D. Originally provision was made for the installation of five rows of superheater tubes instead of the four actually used. The holes which would have been occupied by the fifth row of tubes were closed by rolled-in blind nipples.

The upper superheater box (above the high end of the furnace) served as the inlet and outlet drum. It was constructed with a vertical diaphragm plate located at mid-length inside the box. The box ends were of reduced diameter, threaded, and provided with flanges. The lower superheater box was manufactured with blank ends. Both boxes were provided with small hand hole plates — "bull's eyes" — for inspection, cleaning, and repairs.

BRICKWORK AND INSULATION.

The boiler furnace was constructed as follows:

Floor was insulated with 4 1/2 inch layer of Sil-O-Cel C22 brick laid on the 2 1/2 inch side, covered by two layers of split bricks, Babcock and Wilcox No. 80.

Side walls and both ends of furnace, except immediately around the burners, were made up of a 4 1/2 inch layer of Sil-O-Cel C22 brick as insulation, with Babcock and Wilcox No. 80 firebrick 4 1/2 inches thick; special shape Babcock and Wilcox No. 80 brick were used in the corners of the furnace, covering spaces allowed for expansion.

After the boiler had been installed and a few runs made, additional outer casing plates composed of a thin layer of millboard secured to 1/16 inch iron plate were fitted to all sides of the furnace.

Special shaped tile, backed with insulation, was used between the boiler headers; plastic firebrick and insulation were molded in place around the header ends and superheater boxes, as well as between superheater boxes and cylindrical boxes below them.

INSTALLATION.

The boiler arrived complete with furnace fully bricked and insulated in the manner described above. It was installed in an insulated steel fireroom at the Fuel Oil Testing Plant and fitted with a suitable stack.

A diagram of the steam piping installation by which the boiler was tied into existing lines and condensing system is shown in Figure 5.

During operation at three hundred pounds boiler working pressure steam from the boiler passed into the Plant's 300 pound steam pressure system, pressure being maintained by hand operated bleeder valve. Above 300 pounds steam pressure, it was maintained constant by a different hand controlled bleeder valve (monel metal disc and seat) in main steam line from boiler, the 300 pounds working pressure lines beyond the bleeder being additionally protected by a Crosby safety valve set for 335 pounds lift pressure.

Because of the high temperature superheated steam delivered by this boiler, all auxiliaries except the high pressure feed pump were operated on saturated steam from one of the Plant's other boilers.

Fuel oil was weighed in the Plant's fuel oil weighing tanks, discharging to constant level tank, whence it was pumped to manifold at burners via oil heaters by a motor driven Quimby pump. Pressure was maintained by relief valve returning excess to constant level tank.

All water fed to this boiler was 100 per cent distilled, drawn from a 30,000 gallon water supply tank accurately weighed in the Plant's weighing tanks and discharged into a constant level tank from which feed pumps took suction.

Arrangements were made to feed boiler by either a Worthington three stage high pressure feed pump designed for operation at 750 pounds pressure, or a Worthington 500 G.P.M. pump designed for operation at 450 pounds pressure. The latter was not used when operating at more than 425 pounds pressure. The high pressure three stage pump, which was manufactured expressly for feeding the subject boiler, was driven by a Sturtevant turbine and provided with a Leslie Pressure Regulating Valve, designed to maintain a pump pressure accurate within plus or minus 2 per

DIAGRAM OF STEAM and WATER PIPING INSTALLATION USED IN CONNECTION WITH B & W. "3X" BOILER

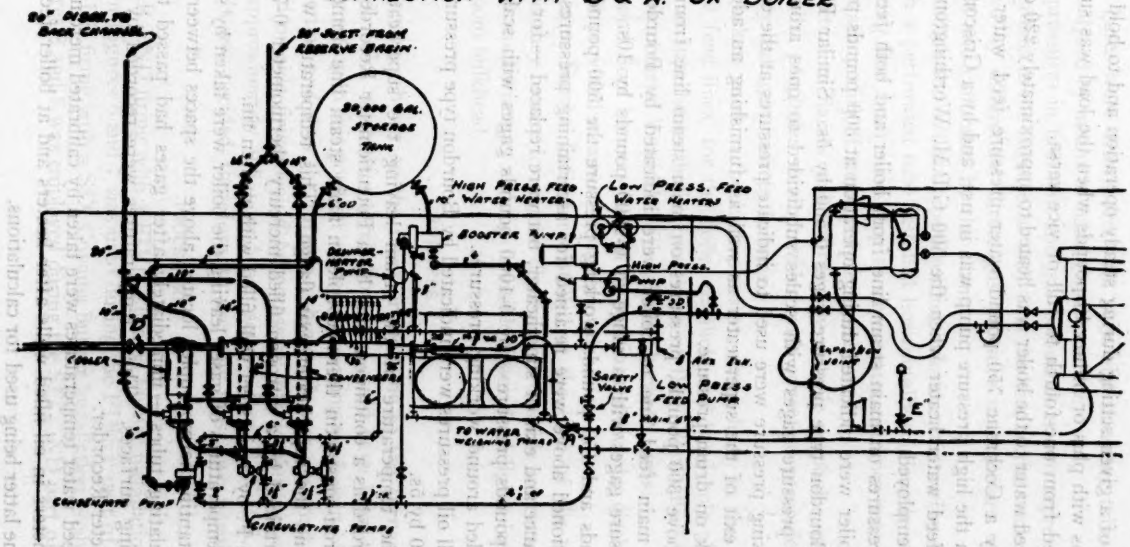


FIGURE 5.

cent of a given setting during steady operation and to hold fluctuations with plus or minus 40 pounds when the load was suddenly varied from one-fourth to full or vice versa.

Feed water to the boiler was heated to approximately 220 degrees F. by a Cochrane 750 pound water pressure feed water heater when the high pressure pump was in use, and by a Griscom Russell feed water heater when the 500 G.P.M. Worthington pump was employed.

Pressures on main steam line from boiler and both feed lines to boiler were indicated during operation at 300 pounds pressure by Bourdon type pressure gages 0-560 by 5s. Similar Bourdon type pressure gages with scales subdivided to ones around the working pressure were used to indicate pressures at the entrance and exit of the superheater as well as furnishing an additional check on drum pressure.

Above 300 pounds, pressures on main steam line from boiler and main feed line to boiler were indicated by Bourdon type pressure gages with a range of 0-1000 pounds by 10s. At 425 pounds and 450 pounds working pressure the 500 pound gages mentioned above were retained for ascertaining pressures at the entrance and exit of superheater, but were replaced — for runs at 600 pounds pressure — by 0-1000 pounds gages with scales subdivided around working pressures.

All oil pressures were indicated by Bourdon type pressure gages 0-500 by 5s.

The temperature of the steam leaving the superheater was recorded as a continuous line by a Potentiometer Recorder using an iron-constantin thermocouple in main steam line immediately beyond the superheater exit flange. This temperature was also ascertained by a nitrogen filled mercury thermometer 0-720 degree F. by 2s in steel well filled with molten tin.

Temperatures of gases leaving the boiler were taken by six iron constantin thermocouples located above the spaces between boiler circulating tubes — immediately after gases had passed the last heating surfaces. These temperatures were recorded by Potentiometer Recorder.

Feed water temperatures were taken by calibrated mercury thermometers in oil-filled wells after heaters and at boiler feed stops — the latter being used for calculations.

Flue gases were drawn from six open end sampling tubes and were analyzed by means of hand Orsat machine.

Air pressures in fireroom and gas pressures at two representative locations in the furnace, as well as at the base of the stack were indicated by differential direct gas pressure manometers manufactured and calibrated by the Plant.

A Babcock and Wilcox type throttling calorimeter drawing steam through a perforated tube installed horizontally midlength of the connecting line from main stop to superheater entrance was used to ascertain quality of steam before superheater.

Inspection of flames was afforded by suitable peepholes in furnace walls; peepholes were provided with covers to preclude unnecessary leakage of air to furnace.

All gaskets used in the high pressure steam lines, in the high pressure feed lines, for the boiler manhole and hand hole plates, as well as superheater bull's eyes, were of the "Flexitallic" type, being monel metal V spirals with asbestos between turns.

Diamond Soot blowers located at positions shown by circles in Figure 1 provided means of soot removal.

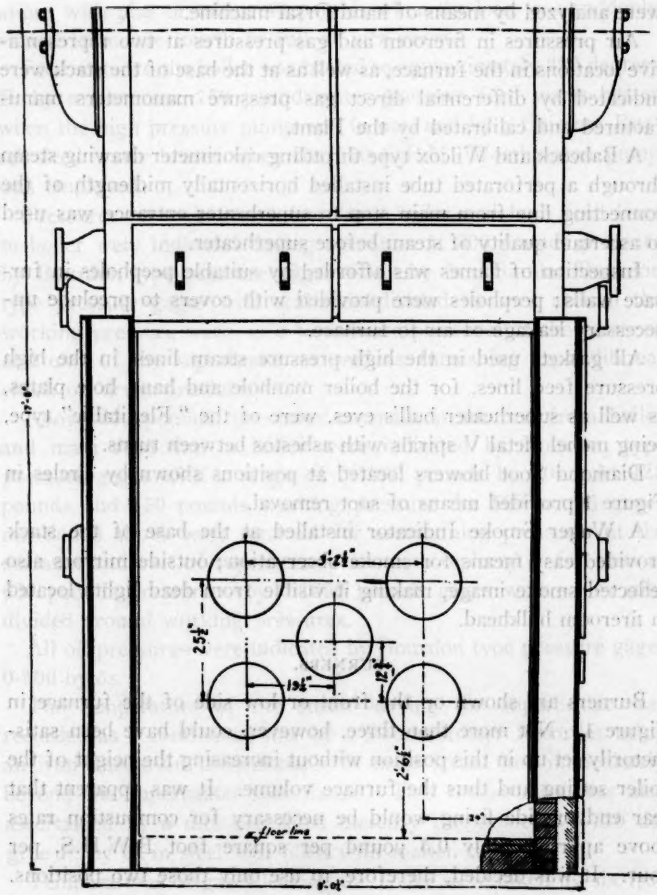
A Wager Smoke Indicator installed at the base of the stack provided easy means for smoke observation; outside mirrors also reflected smoke image, making it visible from dead lights located in fireroom bulkhead.

BURNERS.

Burners are shown on the front or low side of the furnace in Figure 1. Not more than three, however, could have been satisfactorily set up in this position without increasing the height of the boiler setting and thus the furnace volume. It was apparent that rear end or side firing would be necessary for combustion rates above approximately 0.5 pound per square foot B.W.H.S. per hour. It was decided, therefore, to use only those two positions.

PROCEDURE.

REAR END firing with a FIVE Cuyama burner installation — shown in Figures 6 and 7 was used during the FIRST part of this test; SIDE firing with a SIX Cuyama burner installation — shown in Figures 8 and 9 was employed during the SECOND part of the test.



ARRANGEMENT OF BURNERS DURING END FIRING

FIGURE 6.

FIGURE 7.

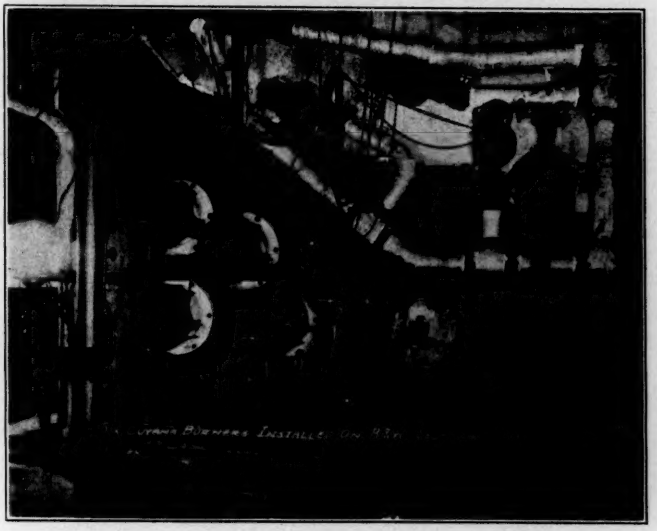


FIGURE 9.

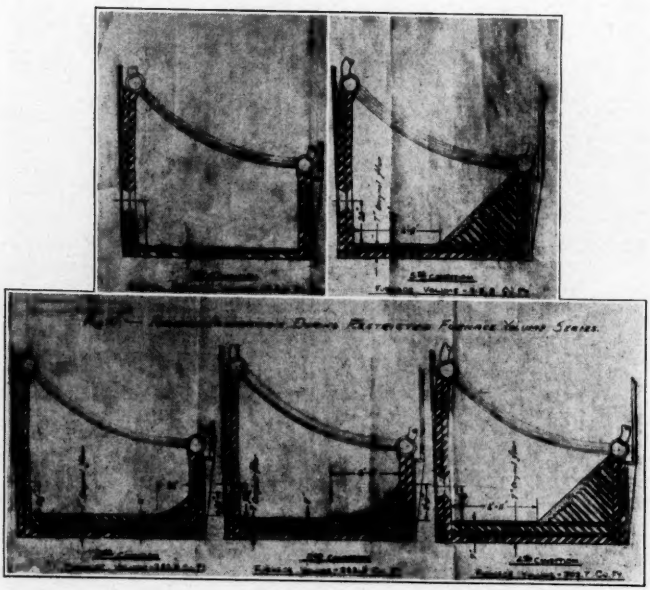
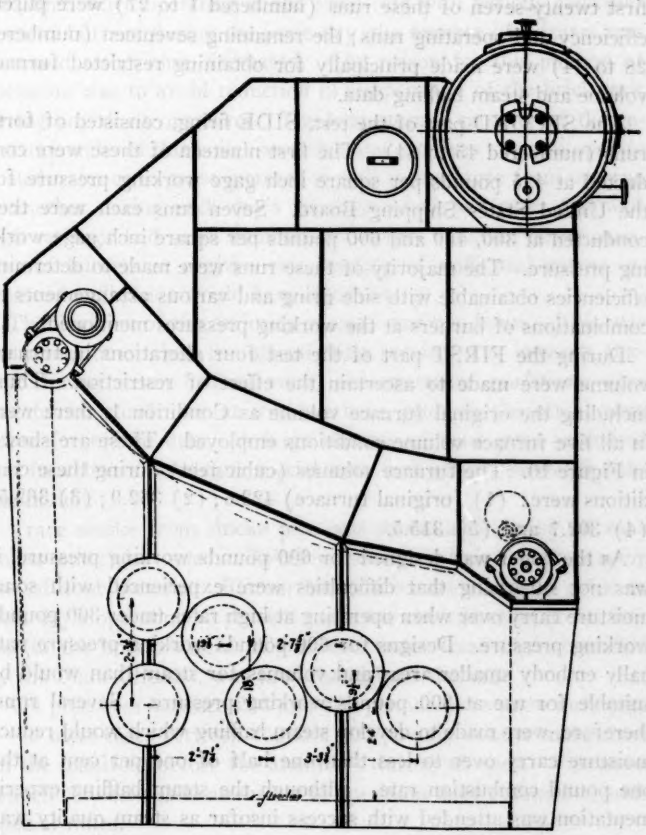


FIGURE 10.



ARRANGEMENT OF BURNERS FOR SIDE FIRING

FIGURE 8.

The FIRST part of the test, REAR END firing, consisted of forty-four runs (numbered 1 to 44), all of which were conducted at 300 pounds per square inch gage boiler working pressure. The first twenty-seven of these runs (numbered 1 to 27) were purely efficiency and operating runs; the remaining seventeen (numbered 28 to 44) were made principally for obtaining restricted furnace volume and steam baffling data.

The SECOND part of the test, SIDE firing, consisted of forty runs (numbered 45 to 84). The first nineteen of these were conducted at 425 pounds per square inch gage working pressure for the United States Shipping Board. Seven runs each were then conducted at 300, 450 and 600 pounds per square inch gage working pressure. The majority of these runs were made to determine efficiencies obtainable with side firing and various arrangements or combinations of burners at the working pressures mentioned.

During the FIRST part of the test four alterations in furnace volume were made to ascertain the effect of restriction. Thus, including the original furnace volume as Condition 1, there were in all five furnace volume conditions employed. These are shown in Figure 10. The furnace volumes (cubic feet) during these conditions were: (1) (original furnace) 423.6; (2) 362.9; (3) 362.5; (4) 302.7 and (5) 315.5.

As the boiler was designed for 600 pounds working pressure, it was not surprising that difficulties were experienced with some moisture carry over when operating at high rates under 300 pounds working pressure. Designs for 600 pounds working pressure natually embody smaller areas and volumes for steam than would be suitable for use at 300 pounds working pressure. Several runs, therefore, were made to develop steam baffling which would reduce moisture carry over to less than one half of one per cent at the one pound combustion rate. Although the steam baffling experimentation was attended with success insofar as steam quality was concerned, the baffling arrangement necessary was cumbersome and further special modification was found necessary to make it more practical for Naval use. Discussion of this work is not included in this article as it is an entirely separate study.

Runs on this boiler were made over a range of combustion rates from slightly less than 0.25 pound per square foot B.W.H.S. per

hour to slightly over 1.0 pound per square foot B.W.H.S. per hour. Operation conducted at different rates of combustion during the same day were always made advancing from a lower to a higher rate, and, when operating at the same rate with various combinations of burners during the same day, the number of burners was increased rather than decreased. The reason for this method of operation was to avoid reduction in temperature of the brickwork during any day's operation. The heat storage capacity of the refractory walls of this boiler was high and care was taken to operate at such rates as to cause increase in wall temperature from one run to another during each day's running — had low rates followed high rates, or had small numbers of burners followed larger numbers of burners at a given rate, cooling of walls and untrue increases in efficiency would have occurred.

The duration of the majority of runs was two hours. This was the actual time taken for official data, but in every instance (except during steam baffling runs) the operating conditions were maintained steady for at least an hour prior to the actual time taken for official data. A large number of runs were continued for four hours to obtain special data and secure thorough substantiation.

Trace smoke from smoke pipe was maintained during all runs unless otherwise noted. Tubes were dusted before each run by use of soot blowers. Bottom blows were administered prior to each day's operation to prevent excessive concentration of impurities in the boiler water. Algor feed water treatment, controlled by the Plant Chemist, was used throughout the test.

Oil pressures between 225 and 300 pounds per square inch gage — in most cases between 250 and 300 pounds — were maintained on the manifold before the atomizers during all runs. Valves on branch connections to atomizers in operation were maintained wide open at times.

The calorimeter was calibrated before the beginning of the test at pressures from 270 pounds to 310 pounds at the entrance to the superheater, and at the close of the series over the range of pressures from 275 to 310, 415 to 460, and 575 to 615 pounds per square inch gage at the superheater entrance.

RESULTS.

Results obtained under END firing conditions at 300 pounds per square inch gage steam working pressure will be first discussed. A tabulation showing principal results obtained under this condition is shown in Figure 11.

At the beginning of this series, runs were made to determine efficiencies and other data obtainable with various combinations of burners and ratios of sprayer plates. It was thought that sprayer plates producing narrower angles of atomized oil spray than those normally used in Naval Express boilers might prove of superior utility because of the arrangement of the burners and the shape of the furnace.

Overall boiler efficiencies at the 0.25 pound combustion rate varied considerably with the combinations of burners and sprayer plates used. They ranged from about 79.7 per cent, when using the two lower registers with narrow spray angles, to 84.6 per cent, when operating the two upper registers with relatively wide spray angles. Repeated check runs at this combustion rate demonstrated that although the flames from the two upper burners were relatively close to the tubes and higher fireroom air pressures therefore were required, the short, fast burning flames thus secured produced higher overall boiler efficiencies than those obtainable by use of the two lower burners. Operation at higher combustion rates definitely established that use of narrow flame angle sprayer plates under these conditions produced results inferior to those obtainable with wide flame angle sprayer plates of Babcock and Wilcox 08 ratio. Therefore, ratio 08 sprayer plates were used exclusively for the remainder of the test.

At the half pound combustion rate overall efficiencies varied considerably with the burner combinations and sprayer plate sizes used; they ranged from 82.2 per cent, when three 3008 sprayer plates were used, to 84.3 per cent with four 3508 sprayer plates in operation. At the three quarter pound combustion rate best results were obtained with four 3008 sprayer plates, overall efficiencies being 83.4 to 83.6 per cent. At the one pound combustion rate, five burners being in operation at about 1000 pounds per burner per hour, overall efficiencies obtained checked at slightly more than 82.7 per cent.

Abridged Data Sheet Showing Pertinent Data During Certain End Working Pressure With Original And Restricted Furnace Volu

Original

1	[illegible]	5	9	17	24	27	10	12	26	11	15	25	14	28	24	26
15	Present furnace volume - cu.ft.	455.4	455.4	455.4	455.4	455.4	455.4	455.4	455.4	455.4	455.4	455.4	455.4	455.4	550.7	550.7
20	Size of square blocks	8x8	8x8	8x10	8x10	8x8	8x8	8x12	8x8	8x8	8x8	8x8	8x8	8x8	8x8	8x8
22	Number of burners	1	4	2	2	2	4	4	4	4	8	4	8	8	2	4
23	Arrangement of burners	2	1-2-4-4	1-2	1-2	1-2	1-2-4-4	1-2-4-4	1-2-4-4	1-2-4-4	All	1-2-4-4	All	All	1-2	1-2-4-4
25	F.O.T.P. Number of Oil.	2-1330	2-1330	2-1330	2-1331	2-1334	2-1330	2-1330	2-1331	2-1330	2-1330	2-1331	2-1330	2-1330	2-1330	2-1330
26	Carbon - Percent	86.90	86.90	86.90	86.99	86.46	86.90	86.90	86.99	86.90	86.90	86.99	86.90	86.88	86.46	86.46
27	Hydrogen - Percent	11.0	11.0	11.0	11.37	11.92	11.0	11.0	11.37	11.0	11.0	11.37	11.0	11.08	11.41	11.41
28	Sulphur - Percent	1.65	1.65	1.65	1.64	1.62	1.65	1.65	1.64	1.65	1.65	1.64	1.65	1.65	1.86	1.86
29	Nitrogen - Percent	0.36	0.36	0.36	0.23	0.16	0.36	0.36	0.23	0.36	0.36	0.23	0.36	0.36	0.28	0.28
30	Oxygen and undetermined - Percent	1.25	1.25	1.25	0.77	0.31	1.25	1.25	0.77	1.25	1.25	0.77	1.25	1.0	0.87	0.87
35	Ash Moisture in Oil	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace
36	Calorific Value B.T.U.	18662	18662	18662	18990	18990	18662	18662	18990	18662	18662	18990	18662	18788	18788	18788
37	Carbon Dioxide - Percent CO_2	13.0	12.1	10.5	11.0	11.2	12.7	13.4	13.2	12.7	12.4	12.4	11.1	12.7	11.2	12.5
38	Oxygen - Percent O_2	5.1	4.9	7.0	6.5	6.2	4.4	4.2	5.1	4.4	5.2	4.1	6.0	5.2	6.1	5.0
39	Carbon Monoxide - Percent CO	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
40	Nitrogen - Percent N_2	81.9	83.0	82.5	82.4	82.5	82.9	82.4	82.7	82.9	82.5	83.5	82.9	82.1	82.7	82.7
51	Press. Outlet Superheater - Lbs. per Sq. In. Abs.	315.2	313.0	311.9	306.1	296.9	713.0	312.6	304.5	314.5	310.7	299.7	304.0	303.0	317.0	314.7
52	Press. Fireside - Lbs. of Water	0.47	1.14	2.30	2.46	2.65	2.90	2.22	2.17	2.42	2.26	2.26	13.45	11.60	2.59	2.27
65	Press. at Burners - Lbs. of Gage	-0.14	-0.12	-0.12	-0.13	-0.12	-0.11	-0.12	-0.14	-0.13	-0.14	-0.14	-0.14	-0.19	-0.17	-0.22
66	Press. of Furnace - Lbs. of Water	0.02	0.06	0.02	0.02	0.04	0.06	0.09	0.06	1.64	1.42	1.22	2.16	2.26	0.04	0.43
76	Temp. of Steam at Superheat Outlet - Average °F.	610.6	622.7	631.6	641.0	639.6	662.2	648.0	661.5	628.0	652.4	661.0	648.5	650.0	654.0	649.0
77	Superheat based on Press. at Outlet - °F.	168.6	203.6	230.4	251.1	219.8	241.8	244.7	242.3	266.8	272.7	268.6	259.0	271.6	251.8	265.6
78	Temp. Feed Water at Feed Stop - °F.	212.9	215.6	217.6	209.5	212.9	227.0	217.0	227.4	221.0	216.7	251.4	206.1	218.2	216.7	219.6
80	Temp. of Gases Leaving Boiler - °F.	496.0	525.0	483.0	482.2	482.2	467.0	467.0	468.2	478.0	473.0	480.0	519.0	527.0	486.0	486.0
91	Temp. of Hot Bulb - °F.	113.0	106.6	117.1	116.5	109.5	103.7	111.0	110.8	110.8	114.0	109.7	123.6	126.1	99.6	91.5
92	Temp. of Wet Bulb - °F.	86.1	84.1	86.8	86.5	80.0	78.0	86.4	77.5	80.5	84.5	77.0	87.6	120.6	77.3	77.2
100	Oil Burned Per Hour - Lbs.	1367	1378.5	1366	1359.5	1352.5	3827	3819	3668.5	3819	3794	3746	4932	4937.0	1356.4	3697.0
101	Oil Burned Per Hour Per Sq.Ft. of Present B.T.S.S. - Lbs.	0.269	0.282	0.269	0.267	0.266	0.497	0.496	0.484	0.721	0.739	0.737	0.974	0.981	0.256	0.265
102	Oil Burned Per Hour Per Sq. Ft. of B.T.S.S. Assuming 19,000 B.T.U. Oil - Lbs.	0.265	0.267	0.266	0.264	0.263	0.488	0.487	0.479	0.737	0.726	0.729	0.955	0.930	0.257	0.261
103	Oil Burned Per Hour Per Cu.Ft. of Furnace Volume - Lbs.	2.99	3.02	2.99	2.98	2.96	5.97	5.98	5.81	9.01	8.86	8.8	11.65	11.77	4.2	8.4
104	Oil Burned Per Hour Per Burner - Lbs.	1367	339.6	683.0	680.0	676.3	631.9	629.7	618.4	954.5	731.0	936.5	990.0	987.4	678.2	678.5
106	Actual Water Evaporated by B.T.S.S. and S.S.S. Per Hour - Lbs.	16894	17366	17361	17181.6	17158	34675	34081	33673.9	31141	29765	30563.9	34990	35210.9	17098.2	34280
107	Quality of Steam at Superheat Inlet - Percent	100	100	100	100	100	99.96	100	100	99.78	99.94	99.87	96.0	99.87	99.98	99.92

END FIRING RUNS AT 300$^{M.A.}$ VOLUMES.

RESTRICTED

	34	35	37	38	39	41	42	43	44
		2000.7	2000.7	2000.7	2000.7	2000.7	2000.7	2000.7	2000.7
		2000	2000	2000	2000	2000	2000	2000	2000
		4	4	4	4	4	4	4	4
		1-0-0-0	1-0-0-0	1-0-0-0	1-0-0-0	All	All	All	All
		0-1000	0-1000	0-1000	0-1000	0-1000	0-1000	0-1000	0-1000
		00.00	00.00	00.00	00.00	00.00	00.00	00.00	00.00
		11.01	11.01	11.01	11.01	11.01	11.01	11.01	11.01
		1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
		0.07	0.07	0.07	0.07	1.0	1.0	0.07	0.07
		Open	Open	Open	Open	Open	Open	Open	Open
		10700	10700	10700	10700	10700	10700	10700	10700
		10.0	10.0	10.1	10.0	10.0	10.7	10.0	10.0
		0.0	0.0	0.0	0.1	0.7	0.0	0.1	0.0
		0	0	0	0	0	0	0	0
		00.7	00.7	00.0	00.0	00.0	00.0	00.0	00.0
		311.0	311.0	313.5	308.4	307.4	304.7	311.0	304.4
		0.00	0.07	0.00	0.72	0.01	10.10	11.70	11.00
		-0.00	-0.10	-0.10	-0.00	-0.00	-0.10	-0.10	-0.00
		0.00	0.00	0.00	1.00	1.70	0.00	0.07	0.07
		000.0	007.0	000.0	000.0	000.0	000.0	000.0	001.0
		000.0	000.0	007.7	000.0	000.0	000.0	000.0	000.1
		000.4	000.0	007.0	000.0	000.0	000.0	001.0	000.1
		000.0	000	000	000	000.0	000	000	000
		00.0	00.0	00.0	00.0	000.0	000.0	000.0	000.0
		77.0	70.0	70.7	00.0	007.0	000.0	07.0	00.1
		2007.0	2070.0	2000.0	2001.0	2001	2004.0	0114.0	2001.0
		0.000	0.007	0.700	0.700	0.000	1.000	1.000	0.000
		0.007	0.001	0.700	0.700	0.770	0.000	0.004	0.004
		0.0	0.0	12.0	12.0	12.70	14.00	14.0	14.01
		000.0	000.0	000.0	000.4	2000.0	2000.0	2000.0	2010.0
		04000	04000.0	00071.0	00074.0	00000	07001.0	07000.0	00070.0
		00.00	00.00	100	00.00	100	00.00	00.00	00.00

ORIGINAL

		8	9	17	36	37	38	39
130	RANGE OF ORDINANCE FOR 3.7-C.S. AND S.A.C.	1.1407	1.1701	1.1707	1.1000	1.1000	1.1000	1.1000
130a	RANGE OF ORDINANCE FOR 3.7-C... AGED	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
131	EQUIVALENT CHARGEMASS FOR 3.7-C.S. AND S.A.C. FOR ZONE - 100.	10000.0	00070.0	00000.0	00010.0	00000.0	00000.0	00000.0
132	EQUIVALENT CHARGEMASS OF 3.7-C.S. AND S.A.C. FOR 10. OF GU. AT TUBE - 1m.	10.000	10.001	10.070	10.000	10.001	10.000	10.000
133	EQUIVALENT CHARGEMASS OF 3.7-C.S. AND S.A.C. FOR 10. OF GU. AGGREGED 10000 C.C.C.	10.000	10.000	10.070	10.001	10.000	10.000	10.000
133a	EQUIVALENT CHARGEMASS OF 3.7-C.S. AGGED FOR 10. OF GU. AT TUBE	10.000	10.001	10.000	10.000	10.000	10.077	10.000
134	RANGE DIMENSIONS	000.0	000.0	007.0	001.0	000.0	1107.0	1100.0
135	PROJECT WEIGHT 3.7-C.S. AGGED	111.0	114.0	117.0	114.0	110.0	000.0	001.0
136	PROJECT WEIGHT AGED A.P.C.	101.0	100.0	100.7	100.7	100.0	010.1	000.0
137	EQUIVALENT CHARGEMASS FOR ZONE FOR 10. OF CHARGE 3.7-C.S. AND S.A.C. - 1m.	0.00	0.00	0.00	0.00	0.00	7.00	0.00
138	PROJECT SPEED AT MID TRAJECTORY, AND EQUIVALENT OF VELOCITY CONSTANT	10.00	00.10	00.10	00.71	07.00	00.00	07.00
139	RANGE 1000: DISTANCE FROM (1 to ZERO ZONE - 000. AT ANGLE OF ELEVATION ZERO 000. AT ZERO ANGLE. - 3.7-C.S. AND PROJECT. (1000-	0.00 1100.0	0.00 1100.1	0.00 1100.0	0.01 1100.0	0.0 1000.1	0.00 1107.0	0.0 1100.0
137	RANGE 1000: ZONE DIFFERENCE IN YARDS, FOR ZERO ZONE - 000. AT ANGLE OF ELEVATION ZERO. AT ZERO ANGLE. - 3.7-C.S. AND PROJECT.	0	0	0	0	0	0	0
138	RANGE 1000: DEVIATION IN YARDS, AND (1- ZERO ZONE - 000. AT ELEVATION ZERO. AT ZERO ANGLE. - 3.7-C.S. AND PROJECT (1000-	0.00 01.1	0.00 00.1	0.00 00.1	0.00 00.0	0.17 00.0	0.00 00.0	0.0 00.1
139	RANGE 1000: ZONE CHANGE, PER DEGREE ZONE (1- ZERO ZONE - 000. AT ELEVATION ZERO. AT ZERO ANGLE. - 3.7-C.S. AND PROJECT	0.00 1010.0	0.00 1000.0	0.00 1001.7	0.00 1000.0	0.00 1007.1	0.00 1007.0	0.0 1000.7
140	RANGE CORRECTIONS TABLES - 3.7-C. AND (1- ZERO.	11.00 0170.0	11.01 0000.0	11.0 0170.7	11.00 0000.1	10.10 0000.7	11.7 0100.0	11.0 0100.0
141	RANGE 1000: ZERO ADJUSTMENT OF GAUGE - 3.7-C. AND PROJECT.	0	0	0	0	0	0	0
142	RANGE 1000: DISTANCE AND DEVIATION CHANGE (1- ZERO ZONE - 000. ZERO ZONE CHANGE PER. - TO 007.000. AT ZERO ANGLE. (1007-	1.10 000.7	1.70 000.7	0.7 010.0	0.0 000.0	0.1 001.0	1.0 000.0	1.0 000.0
143	RANGE 1000: ZERO ZONE, OF GU IN (1- ZERO ZONE - OF 0.0 IN RANGE OF (100- ZERO ZONE. TO 1007. GAUGE LENGTHS 007.	0 0	0.1 2.0	0.01 1.0	0 0.0	0.1 1.0	0.00 11.0	0.00 11.0
144	RANGE 1000: ZERO ZONE. IN ZONE, ZERO DEF. ZONE. AT ZERO ZONE, IN 1007. OF GAUGE LENGTHS ANGLES - 3.7-C. AND PROJECT	0	0	0	0	1	0	0
145	RANGE 1000: RANGE IN YARDS, AND ZONE (1- ZERO. AT ZERO ZONE, IN 1007. OF GAUGE LENGTHS ANGLES - 3.7-C. 3 PROJECTS (1000-	0 0	0.1 0.7	0 0.0	0.0 0.0	0 0.0	0.01 0.0	0.0 0.0
146	RANGE 1000: DEVIATE AND DEVIATION ZONE. AND DEVIATE ZERO ZONE DEF. 1007. AT ZERO ZONE, IN 1007. OF GAUGE LENGTHS ANGLES - 3.7-C. 3 PROJECTS (1000-	0 0	0.1 0.1	0.0 0.0	0.1 1.0	0.1 0.0	0.11 0.0	0.10 0.0
147	RANGE 1000: VELOCITY, PER ZONE ZONE CHANGE, AND DEVIATE ZERO ZONE DEF. 1007. AT ZERO ZONE, IN 1007. OF GAUGE LENGTHS ANGLES - PER 0 ZONE. (1000-	0 0	0.1 0.1	0.00 7.0	0.00 0.0	0.00 0.7	0.00 00.0	0.00 00.0
148	RANGE 1000: DEVIATION IN AND DEVIATION DEVIATED AIR DEVIATION PER - PER 0 0	1.10 1000.0	0.00 0000.0	0.0 000.0	1.0 000.7	1.1 000.1	1.7 000.0	0.0 000.0
149	RANGE 1000 CORRECTED TO 3.7-C.S. S.A.C. (1- AND DEVIATE - 3.7-C. 3 PROJECT (1000-	0.00 1000.0	0.01 000.0	0.0 000.0	0.1 000.0	0.0 000.1	0.0 000.0	0.0 000.0
150	RANGE 1000 CORRECTED TO ZONES AND ZONE (1- 3.7-C. AND PROJECT (1000-	0.00 1000.0	0.01 000.0	0.0 000.0	0.1 000.0	0.0 000.1	0.0 000.0	0.0 000.0
151	RANGE 1000 CORRECTED TO 3.7-C.S. AND (1- S.A.C., 3.7-C. AND PROJECT (1000-	0 0	0.1 0.1	0.07 12.7	0.00 7.0	0.07 12.7	0.00 100.0	0.7 100.0
152	RANGE ASSIGNED TO SMALL INCREMENT 3.7-C. PER 10. OF GU. - 3.7-C.	10000.0	10000.0	10777.1	10000.0	10000.1	10000.1	10000.0
153	RANGE ASSIGNED TO 3.7-C.S. PER 10. OF GU. - 3.7-C.	10000.0	10000.0	10177.1	10077.0	10000.1	10000.1	10000.0
154	RANGE ASSIGNED TO 3.7-C. PER 10. OF GU - 3.7-C.	1000.0	1001.0	1077.7	1000.1	1000.0	1000.0	1000.0
155	RANGE ADJUSTED AND CORRECTED TO 3.7-C.S. 3 S.A. PER 10. OF GU. - 3.7-C.	10000.0	10000.1	10000.0	10010.0	10000.7	10000.0	10000.0
156	OVERALL DEV. OF 3.7-C.S., S.A.C., PROJECT AND ZONE - ZONES	00.00	00.00	00.00	01.07	01.00	00.00	00.00
157	OVERALL DEVIATION OF 3.7-C.S. AND S.A.C. PROJECT	100	00.00	00.00	00.00	00.01	00.00	00.00
158	OVERALL DEVIATION OF ZONES AND ZONES - ZONES	00.00	00.00	00.00	00.00	00.00	00.07	00.00
159	OVERALL DEVIATION OF 3.7-C.S., S.A.C. PROJECT AND ZONES ALLOCATED TO ZONES	00.00	00.00	00.00	00.00	00.00	00.00	00.00
160	ZONES DEVIATION - ZONES	77.00	00.77	00.00	01.00	01.00	01.00	01.00
161	ZONE - ZAC. DEV. TRIAL, ZONES	00.00	00.00	00.00	00.00	00.00	00.00	00.00

FIGURE 11.

END FIRING RUNS AT 300# VOLUMES.

RESTRICTED

ORIGINAL

		A	B	M	N	O	P	Q	R	S
110	RATIO OF EXPANSION FOR L.P.C.C. AND H.P.C.	1.1287	1.1781	1.1749	1.1865	1.1899	1.1919	1.1894	1.1907	1.1885
110a	RATIO OF EXPANSION FOR L.P.C.C. ENGINE	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
111	CUBICAL CONTENTS FOR L.P.C.C. AND H.P.C. FOR ENDS - CUS.	10965.4	10974.1	10966.9	10414.0	10468.0	10466.8	10481.0	10495.6	10384.
112	CUBICAL CONTENTS OF L.P.C.C. AND H.P.C. FOR LE. OF CYL. AT PIST. - CUS.	15.999	15.991	15.979	14.888	14.891	14.913	14.918	14.924	14.979
113	CUBICAL CONTENTS OF L.P.C.C. AND H.P.C. FOR CL. OF CYL. CLEARANCE SPACE C.C.	15.979	14.983	14.974	14.891	14.893	14.894	14.920	14.927	14.980
113a	CUBICAL CONTENTS OF H.P.C.C. CLEAR. FOR CL. OF CYL. AT PIST.	15.999	14.994	14.988	14.890	14.894	14.871	14.920	14.930	15.004
114	CUBICAL CONTENTS	886.5	890.5	897.4	881.0	888.3	1197.0	1190.9	1162.8	1779.
115	PISTON SPEED	111.0	114.1	117.2	114.4	116.0	224.6	221.0	224.0	224.
		111.0	114.1	116.7	115.7	116.9	213.1	210.0	217.4	217.
117	EXPANSION CONSTANTS FOR ENDS FOR CL. OF CYLINDER L.P.C.C. AND H.P.C. - Ibs.	8.49	8.64	8.60	8.68	8.69	7.88	7.88	7.16	10.
121	PRESSURE DROPS AND RATIO MECHANICAL AND PRESSURE OF CYLINDER CONDENSATION	19.89	20.16	19.13	19.71	27.44	18.89	17.89	19.89	18.
126	HEAT LOSS: RADIATION FROM Ib. OF FUEL FIRED - BTU. AT BOILER OR SUPERH. - H.P.C. (BTU)	8.83	8.83	8.80	8.81	8.4	8.80	8.8	8.3	8.8
		1138.6	1138.1	1138.6	1178.8	1210.1	1137.4	1138.9	1178.8	1138.
127	HEAT LOSS: DUE MOISTURE IN FUEL FIRED - BTU. AT BOILER OR SUPERHEATER - H.P.C. (BTU)	0	0	0	0	0	0	0	0	0
128	HEAT LOSS: SUPERHEAT IN STEAM. AND (S. FUEL FIRED) - BTU. AT SUPERHEATER - H.P.C. (BTU)	0.84	0.85	0.84	0.84	0.17	0.80	0.8	0.8	0.8
		41.4	42.4	42.4	44.8	39.8	39.8	41.1	42.8	41.
129	HEAT LOSS: DUE UNBURNT GAS IN FUEL FIRED (%) - BTU. AT BOILER OR SUPERHEATER - H.P.C. (BTU)	8.83	8.86	8.85	8.81	8.80	8.81	8.8	8.8	8.8
		1218.5	1224.9	999.7	1065.8	1047.1	1038.5	1065.7	1020.1	1039.
130	TOTAL UNACCOUNTED LOSSES - B.T.U. AND (% FUEL)	11.08	11.43	11.6	11.05	12.10	11.7	11.8	11.9	11.7
		1530.5	1580.4	1575.7	1586.1	1588.7	1592.9	1596.6	1601.8	1597.
131	HEAT LOSS: RADIAN ABSORBED BY GASES - B.T.U. AND (%)	0	0	0	0	0	0	0	0	0
132	HEAT LOSS: RADIAN AND CONVECTION CARRIED (% FUEL) AWAY FROM DRY FURNACE WALL - BTU. AT BOILER OR SUPERH. - H.P.C.	1.14	1.74	2.7	2.8	2.1	1.6	1.8	1.1	1.8
		158.4	236.7	315.2	428.2	281.2	216.7	242.6	151.7	235.
133	HEAT LOSS: RADIATION AND CONVECTION CARRIED (% FUEL) FROM DRY FURNACE WALL - BTU. AT BOILER OR SUPERH. - H.P.C.	0	0.1	0.01	0	0.1	0.06	0.05	0.07	0.1
		0	2.3	1.0	0.4	1.0	11.6	11.0	12.8	12.
134	HEAT LOSS: DUE UNBURNT GAS IN FUEL FIRED (%) - BTU. AT BOILER OR SUPERH. - H.P.C.	0	0.1	0	0.1	0	0.01	0.0	0.01	0.0
		0	0.7	0.0	0.1	0.0	1.4	0.0	2.0	1.
135	HEAT LOSS: SENSIBLE AND LATENT HEAT IN MOISTURE FROM H2 IN FUEL (%) - BTU. AT BOILER OR SUPERH. - H.P.C. AND (BTU)	0	0.1	0.1	0.1	0.2	0.11	0.10	0.1	0.1
		0	2.1	2.0	1.9	2.4	22.0	21.0	19.0	21.
137	HEAT LOSS: UNKNOWN, NOT KNOWN FROM KNOWN, AND UNACCOUNTED FOR - BTU % FUEL - H.P.C. AND (BTU)	0	0.1	0.06	0.08	0.08	0.06	0.08	0.2	0.07
		0	10.1	7.6	6.3	8.7	84.0	84.0	84.7	100.
140	HEAT LOSS: UNACCOUNTED FOR AND Difference between Unaccounted for - BTU & %	7.10	5.40	6.3	1.8	1.1	1.9	1.8	1.3	1.78
		1000.9	640.4	800.9	304.7	203.1	320.4	304.8	220.6	303.
139	HEAT LOSS CHARGEABLE TO L.P.C.C. H.P.C. (%) AND ENGINE - B.T.U. & ENGINE (BTU)	0.85	0.84	0.7	0.1	0.8	0.8	0.6	0.1	0.80
		1020.8	990.3	882.3	778.8	882.1	936.0	926.8	960.8	975.
138	HEAT LOSS CHARGEABLE TO FRICTION AND ENGINE - B.T.U. AND (BTU)	0.85	0.81	0.6	0.1	0.8	0.8	0.8	0.4	0.5
		1020.8	972.1	870.8	743.8	864.6	914.6	921.8	888.1	824.
141	HEAT LOSS CHARGEABLE TO L.P.C.C. AND H.P.C., L.P.C.C. AND H.P.C. AND (BTU)	0	0.1	0.07	0.01	0.07	0.06	0.7	0.7	1.07
		0	10.1	12.7	7.0	13.7	119.4	120.8	128.7	200.
142	HEAT ABSORBED BY WATER JACKETING L.P. PER LB. OF GAS - B.T.U.	14068.0	13966.4	13799.1	13868.0	13895.1	13883.1	13868.0	13886.0	13860.
143	HEAT ABSORBED BY H.P.C.C. PER LB. OF GAS - B.T.U.	13896.0	13899.0	14117.4	14070.0	14305.4	13918.6	13908.0	13886.0	13851.
144	HEAT ABSORBED BY L.P. PER LB. OF GAS - B.T.U.	1488.0	1891.0	1892.7	1732.1	1787.0	1863.0	1868.0	1857.1	1886.
145	HEAT AVAILABLE FOR ABSORPTION BY L.P.C.C. & H.P. PER LB. OF GAS - B.T.U.	14848.3	14886.5	15089.8	15811.9	14864.9	15851.0	15776.3	15868.7	15866.
146	OVERALL EFF. OF L.P.C.C., H.P.C., FRICTION AND ENGINE - PERCENT	88.87	88.88	84.68	84.87	84.62	84.30	88.88	84.83	83.
147	COMBINED EFFICIENCY OF L.P.C.C. and H.P.C. - PERCENT	100	99.84	99.82	99.92	99.71	99.92	99.88	99.87	99.
148	COMBINED EFFICIENCY OF FRICTION AND ENGINE - PERCENT	90.66	94.69	95.88	95.89	94.84	94.87	94.74	96.17	96.
150	COMBINED EFFICIENCY OF L.P.C.C., H.P.C., FRICTION AND ENGINE ALSOCIATED WITH ASSOCIATED LOSSES	90.66	94.88	95.81	95.89	94.84	94.87	94.94	96.17	96.
151	FRICTION EFFICIENCY - PERCENT	77.86	80.72	82.16	81.04	81.80	81.88	81.86	81.89	80.
152	IND. - ENG. EFF. CURVE, APPLIED	88.31	88.30	88.40	88.17	88.42	88.28	88.35	88.13	88.

	25	27	28	29	31	33	35	39
	312.7	312.7	312.7	312.5	312.5	312.5	312.7	312.5
	2000	2000	2000	2000	2000	2000	2000	2000
	4	4	4	4	8	8	8	8
	1-2-4-8	1-2-4-8	1-2-4-8	1-2-4-8	All	All	All	All
	8-1295	8-1295	8-1295	8-1295	8-1295	8-1295	8-1295	8-1295
	88.40	88.40	88.40	88.40	88.88	88.88	88.40	88.40
	11.31	11.31	11.31	11.31	11.26	11.26	11.31	11.31
	1.96	1.96	1.96	1.96	1.99	1.99	1.96	1.96
	0.39	0.39	0.39	0.39	0.35	0.35	0.39	0.39
	0.97	0.97	0.97	0.97	1.0	1.0	0.97	0.97
	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace
	10788	10788	10788	10788	10788	10788	10788	10788
	15.5	15.9	15.1	12.6	15.5	15.7	15.6	15.9
	5.9	5.0	5.4	4.1	5.7	5.5	5.1	5.8
	0	0	0	0	0	0	0	0
	83.7	83.7	83.5	83.5	83.3	84.0	83.8	83.3
	316.7	311.0	312.5	308.4	307.4	304.7	311.0	304.0
	5.07	5.46	5.71	9.01	15.16	11.78	11.09	11.95
	-0.22	-0.19	-0.19	-0.29	-0.30	-0.15	-0.19	-0.25
	0.43	0.52	1.65	1.76	2.89	2.57	2.57	2.73
	663.0	667.0	668.0	663.0	668.0	661.0	668.0	661.0
	265.5	266.0	267.7	265.0	270.0	278.0	272.0	272.1
	219.4	223.1	217.0	227.0	218.0	225.0	221.0	220.1
	484.0	488	485	492	488.0	515	500	505
	91.5	95.4	94.0	100.0	116.5	116.5	100.5	106.5
	77.2	78.0	77.7	80.2	107.0	112.0	87.5	94.1
	2227.0	2275.2	2230.5	2221.9	2221	2104.5	2116.0	2141.5
	0.365	0.397	0.782	0.782	0.994	1.004	1.004	0.994
	0.497	0.501	0.745	0.745	0.970	0.989	0.994	0.994
	8.4	8.5	12.6	12.5	15.78	14.06	14.9	14.01
	680.5	686.0	686.6	686.4	2000.2	2032.9	2020.5	2012.5
	94280	94288.5	89871.8	89874.0	88809	87391.9	87885.9	88878.0
	99.92	100	99.93	100	99.85	99.76	99.85	99.78

FIGURE 11.

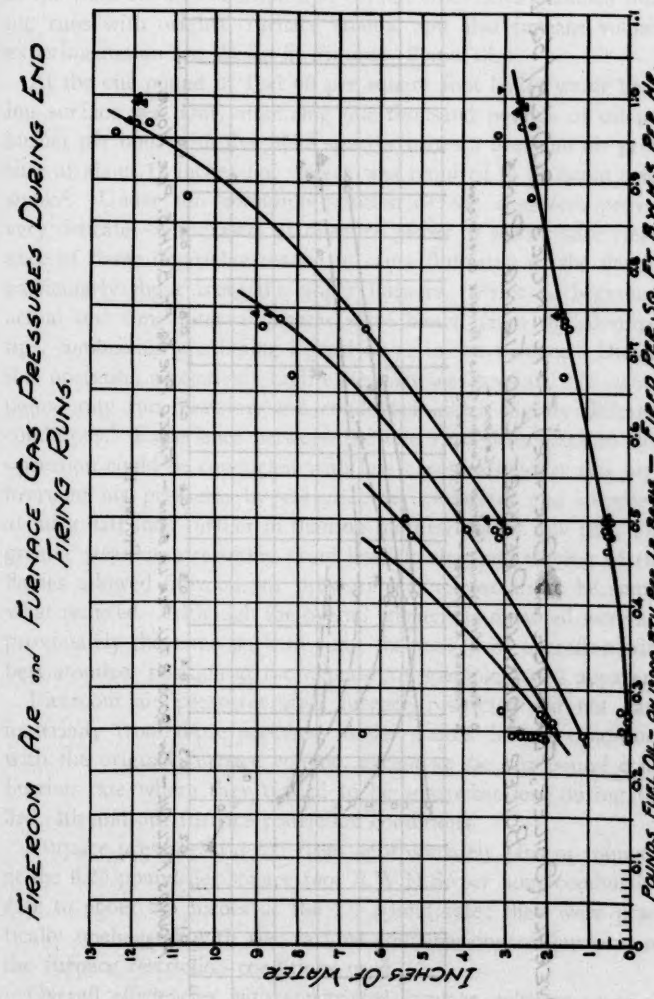

FIREROOM AIR and FURNACE GAS PRESSURES DURING END FIRING RUNS.
INCHES OF WATER
POUNDS FUEL OIL - OR 19,000 BTU PER LB. BASIS - FIRED PER SQ. FT. B.W.H.S. PER HR.

FIGURE 12.

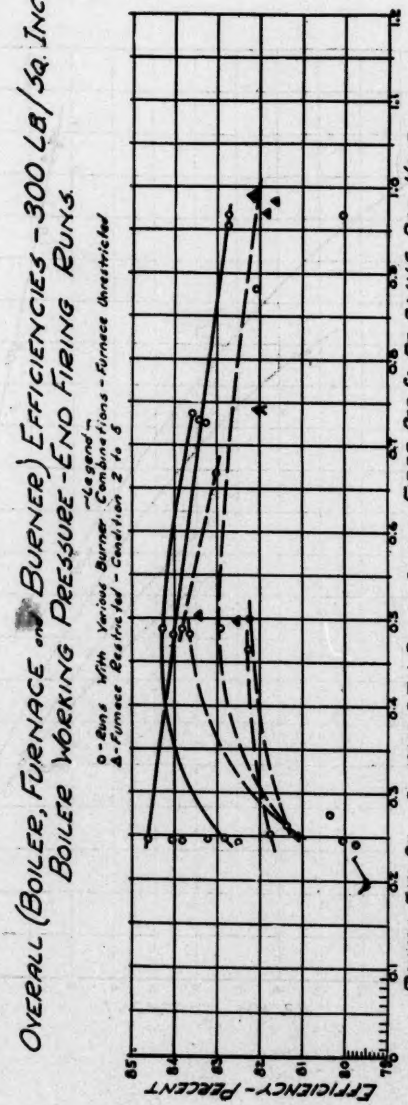

OVERALL (BOILER, FURNACE AND BURNER) EFFICIENCIES - 300 LB/SQ. INCH BOILER WORKING PRESSURE - END FIRING RUNS.
Legend -
O - Runs With Various Burner Combinations - Furnace Unrestricted
Δ - Furnace Restricted - Condition 2 to 5
EFFICIENCY - PERCENT
85
84
83
82
81
80
79
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
POUNDS FUEL OIL - ON 18000 B.T.U. PER LB. BASIS - FIRED PER SQ. FT. B.H.H.S. PER HOUR.

FIGURE 13.

Curves showing fireroom air pressure necessary for operation at the trace smoke condition and overall efficiencies obtained during runs with original furnace volume and also furnace volume experimentation are shown in Figures 12 and 13.

At the one pound of fuel oil per square foot boiler water heating surface per hour, atomizing one thousand pounds of oil per burner per hour with five 3008 sprayer plates a fireroom air pressure of about 12 inches (of water) was required to maintain trace smoke. Under this condition position of the atomizers proved very delicate—withdrawal of distance pieces to secure bare clearance of flame invariably tended to cause fluttering of the flames, particularly those from the upper burners. Prior to beginning actual test time interval, flames were blown from the atomizer tips, combustion continuing a yard or so in the furnace. During this operation a relatively high excess air was present. A second pound rate run however, was conducted under slightly different conditions. Experience between the two runs demonstrated that operation could be conducted with less excess air under this high fireroom air pressure, by carrying the atomizers and impellers about a half inch further in than on the first pound rate run. The greater steadiness resulting from less blowing and tearing of the flames allowed fireroom air pressure and excess air to be somewhat reduced. Although the overall efficiencies obtained were approximately the same on both runs, the fallacy of operation with best atomizer position at the expense of steadiness was apparent.

Fireroom air pressures with furnace restricted did not vary materially from those necessary under similar burner conditions with the original furnace volume, except at the one pound combustion rate where they tended to be somewhat less during the 3rd, 4th and 5th furnace restriction conditions.

Furnace pressures ranged from approximately plus or minus 0 at the 0.25 pound per square foot B.W.H.S. per hour combustion rate to about 2.5 inches at the 1.0 pound rate; they were practically unchanged with the various combinations of burners, or the furnace restriction conditions used.

Overall efficiencies with constricted furnace volumes were all somewhat lower than those made under similar conditions with

the original furnace volume. At the one pound per square foot B.W.H.S. per hour combustion rate these efficiencies were:

Furnace Condition	Furnace Volume Cubic Feet	B. T. U. Fired Per Cubic Foot per Hour	Efficiencies
			Per Cent
3	362.5	263,606	82.2
4	302.7	317,179	82.1
2	362.9	257,989	81.8
5	315.5	300,476	81.6

It is to be noted that although furnace volume was altered between a minimum volume of 302.7 and a maximum of 362.9 cubic feet, overall efficiency did not vary more than 0.6 per cent, for all practical purposes overall efficiency could be called 81.9 per cent at the 1.0 pound combustion rate for all four restricted furnace conditions. This average, however, was over 1 per cent below the efficiencies obtained with unrestricted furnace—423.6 cubic foot furnace volume.

It is believed, however, that had the runs with restricted furnace volume been conducted for periods of several hours, efficiencies more closely approaching — possibly even above — those with unrestricted furnace would have been obtained. The large volume of brickwork comprising the restrictions never became thoroughly heat soaked and undoubtedly continued to absorb heat throughout the runs.

Runs at the 0.25, 0.50 and 0.75 pound per square foot B.W.H.S. per hour combustion rates were also made with furnace condition 4, furnace volume 302.7 cubic feet, as well as at the 0.75 rate with furnace condition 5—furnace volume 315.5 cubic feet. In each case the overall efficiencies obtained were materially below those during similar runs without furnace restriction. This was not due to any marked change in combustion conditions, however, but to the large volume of brickwork present. At these rates an even longer period for heat soaking would have been required than at the 1.0 pound rate, but it is believed that, had it been possible to conduct these runs for sufficiently long periods,

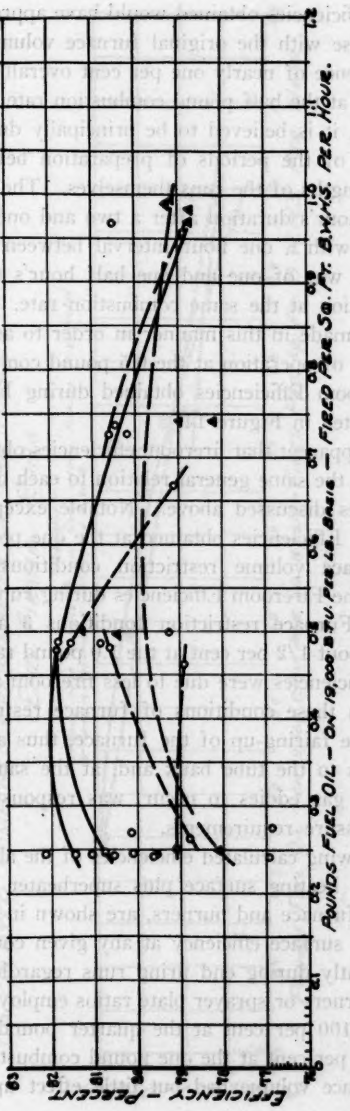

FIREROOM EFFICIENCIES DURING END FIRING RUNS.
—Legend—
O - Runs with various Burner Combinations - Furnace unrestricted
Δ - Runs with various Burner Combinations - Furnace Restricted: Condition as in O
EFFICIENCY - PERCENT
POUNDS FUEL OIL - OR 19,000 B.T.U. PER LB. BASIS - FIRED PER SQ. FT. B.W.H.S. PER HOUR.

FIGURE 14.

the overall efficiencies obtained would have approached or possibly exceeded those with the original furnace volume.

The difference of nearly one per cent overall efficiency between the two runs at the half pound combustion rate with furnace condition No. 4, it is believed to be principally due to difference in the duration of the periods of preparation before the runs and also to the lengths of the runs themselves. The first of these two was of one hour's duration after a two and one half hour run at the quarter, with a one hour interval between runs; the second of these two was of one and one half hour's duration after six hours' operation at the same combustion rate. These runs were intentionally made in this manner in order to ascertain difference due to length of operation at the 0.5 pound combustion rate.

The Fireroom Efficiencies obtained during END firing conditions are plotted in Figure 14.

It will be apparent that fireroom efficiencies obtained are, in general, in about the same general relation to each other as the Overall Efficiencies discussed above. Notable exceptions to this are the Fireroom Efficiencies obtained at the one pound rate with the various furnace volume restriction conditions. They average higher than the Fireroom Efficiencies during runs without furnace restriction. Furnace restriction conditions 3 and 4 allowed an increase of about 1/2 per cent at the 1.0 pound rate. These higher Fireroom Efficiencies were due to less fireroom air pressure being required with these conditions of furnace restriction. It is believed that the fairing-up of the furnace, thus allowing an easier flow of gases to the tube bank and, at the same time, reducing tendency for gas eddies to occur, was responsible for less fireroom air pressure requirements.

Curves showing calculated efficiencies of the absorption surfaces (boiler water heating surface plus superheater heating surface), efficiency of furnace and burners, are shown in Figure 15.

Absorption surface efficiency at any given combustion rate did not vary greatly during end firing runs regardless of the combinations of burners or sprayer plate ratios employed. They ranged from nearly 100 per cent at the quarter pound combustion rate to about 96.8 per cent at the one pound combustion rate. Alterations in furnace volume had but little effect upon the efficiency

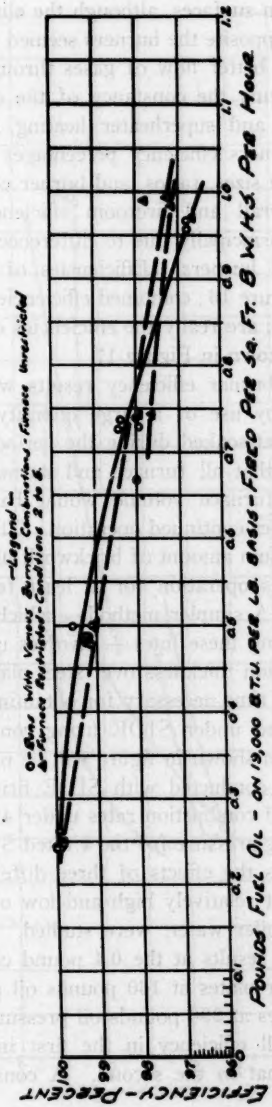

CALCULATED COMBINED EFFICIENCIES OF THE ABSORPTION SURFACES DURING END FIRING RUNS
-Legend-
O-Runs With Vanes Burner Combustion - Furnace Unrestricted
Δ-Furnace Restricted - Conditions 2 to 6.
EFFICIENCY - PERCENT
POUNDS FUEL OIL - ON 19,000 B.T.U PER LB. BASIS - FIRED PER SQ. FT. B.W.H.S. PER HOUR.

FIGURE 15.

of the absorption surfaces, although the elimination of the corner of the furnace opposite the burners seemed to cause an increase — probably due to better flow of gases through the boiler.

Far from having the constancy of the curve for boiler water heating surface and superheater heating surface efficiency, the furnace and burners efficiency percentages varied markedly with the sprayer plate sizes, ratios, and burner combinations employed. Changes in overall and fireroom efficiencies — previously discussed — were principally due to differences in the efficiencies of the furnace and burners. Efficiencies of furnace and burners are shown in Figure 16; combined efficiencies eliminating unavoidable losses, which are really the efficiencies on a 100 per cent possible basis, are shown in Figure 17.

Furnace and burner efficiency results with restricted furnace were befogged by use of a large quantity of brickwork which never became heat-soaked during the periods of operation. It is believed certain that all furnace and burner efficiencies obtained with restricted furnace volume would have been substantially increased by longer continued operation. The thorough heat soaking of the maximum amount of brickwork used would have necessitated continuous operation for at least forty-eight hours prior to data taking. A simpler method — which will be employed in further work along these lines — involves use of standard brickwork and insulation thickness over steel plate false bottoms, thus reducing running time necessary for obtaining more accurate data.

Results obtained under SIDE firing conditions using the six burner installation shown in figure will be next discussed.

The first runs conducted with SIDE firing were made at the 0.4 and 0.5 pound combustion rates under 425 pounds per square inch gage working pressure for the United States Shipping Board. During this series the effects of three different burner combinations, operation at relatively high and low oil pressures and high salt content in boiler water, were studied.

Comparison of results at the 0.4 pound combustion rate using four 3508 sprayer plates at 160 pounds oil pressure against four 3908 sprayer plates at 290 pounds oil pressure confirmed previous experience, overall efficiency in the first instance being over 4 per cent below that in the second. A considerable quantity of

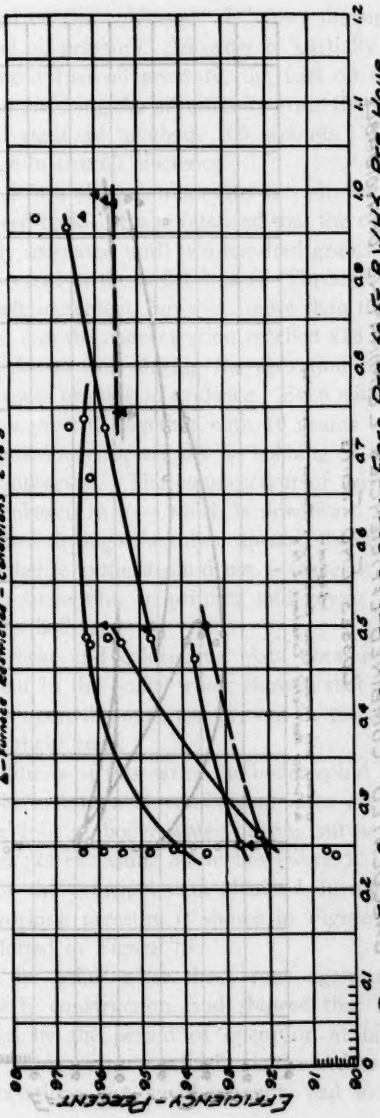

CALCULATED EFFICIENCIES OF FURNACE AND BURNERS DURING END FIRING RUNS.
Legend
EFFICIENCY - PERCENT
POUNDS FUEL OIL - ON 19,000 B.T.U. PER LB. BASIS - FIRED PER SQ. FT. B.W.H.S. PER HOUR.

FIGURE 16.

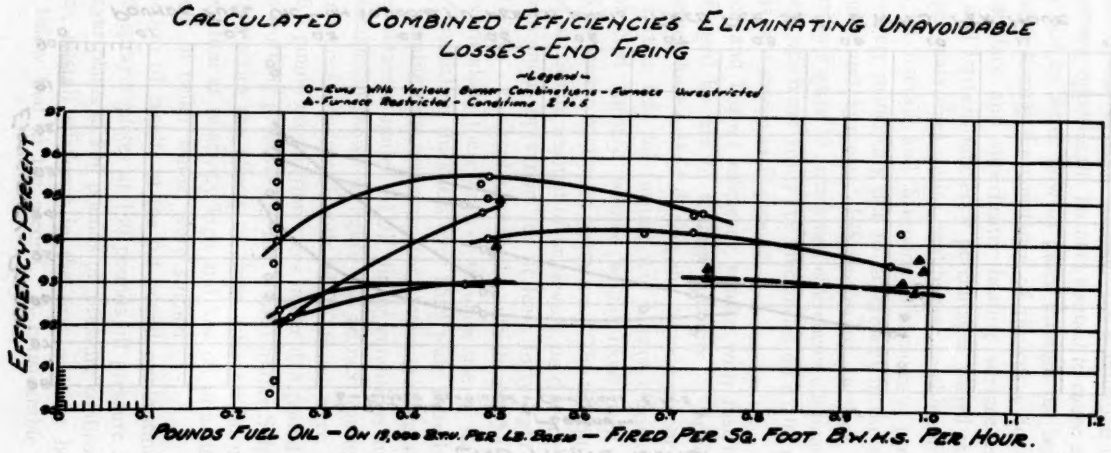

CALCULATED COMBINED EFFICIENCIES ELIMINATING UNAVOIDABLE LOSSES-END FIRING
-Legend-
O-Ends With Various Burner Combinations-Furnace Unrestricted
Δ-Furnace Restricted-Conditions 2 to 6
EFFICIENCY-PERCENT
POUNDS FUEL OIL - ON 19,000 B.T.U. PER LB. BASIS - FIRED PER SQ. FOOT B.W.H.S. PER HOUR.

FIGURE 17.

residue also lodged on the "A" and "B" rows during operation at 160 pounds fuel oil pressure. In order to partially counteract the effect of reduced fuel oil pressure, the fuel oil temperature was increased, thus reducing the oil viscosity from the 150 seconds Saybolt Universal standard to about 100 seconds. This caused little if any change in overall efficiency.

A condenser leak was then simulated by introducing unpurified sea salt into the feed tank. It was intended that the concentration would be gradually increased until two hundred grains of chlorine per gallon were present in the boiler water. Through an error in making the last salt increment, however, more than the necessary amount was added and the concentration reached 273 grains chlorine per gallon. At no time during the operation of the boiler at these conditions was priming in evidence. Even with 273 grains chlorine per gallon present, together with 16 grains of sulphates and 14 grains of alkalinity, practically no bobbing of water in the gage glasses was observed. The construction of this boiler and the course of circulation in it — which is downward through the vertical headers, then through the tubes, upward through the diagonal headers, and thence to the steam drum — makes it to a considerable degree less susceptible to priming and geyser action than "A" type Express boilers.

Study of superheat and calorimeter data obtained with and without salt present in the boiler water showed that salinity had little if any effect upon the moisture present in the steam under the conditions of these runs.

Under the conditions of this series the boiler could be operated at the trace smoke condition at combustion rates up to the half pound per square foot of boiler water heating surface with fireroom air pressures not exceeding 3.5 inches (water).

A tabulation of the principal data obtained during this series at 425 pounds working pressure is shown in Figure 18; overall efficiencies are plotted in Figure 19.

Inspection of the boiler after these runs again revealed its exceptionally sturdy construction, and showed that it had been entirely unharmed by the period of operation at high salinity. The superheater tubes were practically entirely free from salt, contrary to expectations, but small amounts had lodged in the

closed-end nipples which had been installed by the manufacturer in place of the fifth row superheater tubes. A demonstration of cleaning water sides of boiler tubes and the steam sides of superheater tubes was held. Tube No. 18 " A" row, which had a slight salt exudation around the end at the lower box, was rerolled. A large number of superheater tubes were rerolled as a demonstration of the facility with which this operation could be accomplished. One of the superheater closed end nipples was removed

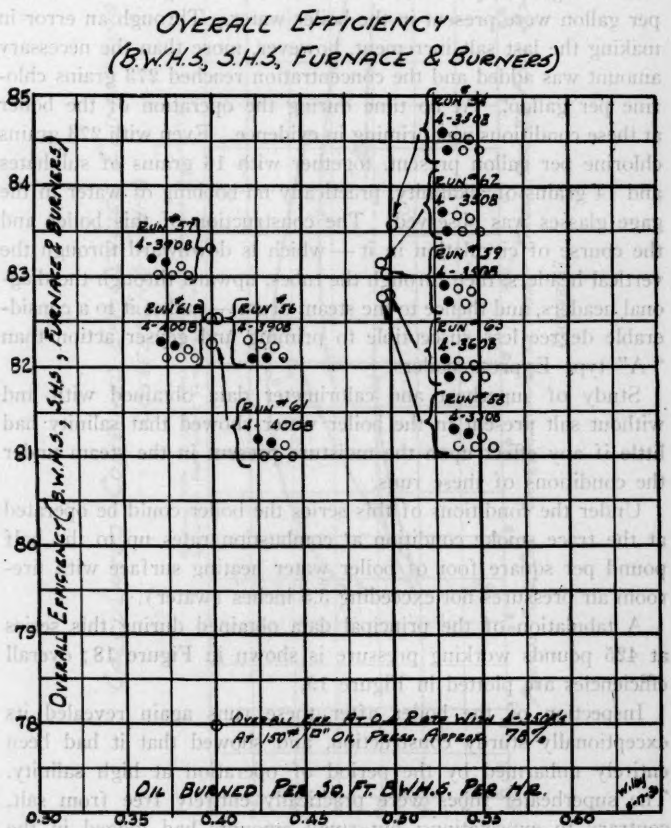


FIGURE 19.

Form No. W—A-1-25—1,000

FUEL OIL TE
BOILER TE
Test No.

Date

ITEM No.	ITEM
3	Boiler Water Heating Surface, Sq. Ft.
18	Type of Registers
19	Type of Atomizer
20	Size of Sprayer Piston, Tips & Plugs
23	Number of Burners
29	Arrangement of Burners
36	Calorific Value of Fuel, BTU
37	Carbon Dioxide (CO_2) %
38	Oxygen (O_2) %
39	Carbon Monoxide (CO) %
40	Nitrogen (N_2) %
53	Pressure Below Stack Damper—Lbs. Sq. In. Gauge
54	Pressure—Steam at Blower Turbine Governor Valve—Lbs. Sq. In. Gauge
61	Air Pressure, Barometric, Ins. of Mercury
62	Air Pressure, Fire Room, Ins. of Water
63	Pressure at Beginning of Smoke Pipe Ins. of Water
64	Pressure in Furnace, Ins. of Water
67	Temp. Calorimeter S.H. Inlet S. °F
74	Temperature Superheater Outlet °F (Average Temp. of Right & Left Le
77	Superheat Based on Temp at the "T" °F
79	Temperature—Feed Water at Boiler
81	Temperature—Dry Bulb °F
82	Temperature—Wet Bulb °F
87	Duration of Test—Hours
90	Oil Burned—Total
92	Actual Total Evaporation of F.W.H. Surface & Superheater—Lbs.
93a	Oil Burned Per Hour—Lbs.
94	Oil Burned Per Hour Per Sq. Ft. Boiler Water Heating Surface Lbs.
95	Oil Burned Per Horse Per Hour
96	Actual Water Evaporated by Water Heating Surface & Superheater Per Hour—Lbs.
110	Factor of Evaporation for Boiler water Heating Surface & Superheater
111	Equivalent Evaporation of Boiler Wa & Superheater Heating Surface Per Hr.—Lbs.
112	Equivalent Evaporation of Boiler Wate & Superheater Heating Surface Pe Lb. of Oil as Fired—Lbs.
113	Heat Absorbed by the Actual Evaporation Per Lb. of Oil—B.T.
114	Overall Efficiency of Boiler Water Heating Surface & Superheater Furnace & Grate—%

FUEL OIL TESTING PLANT, NAVY YARD, PHILA., PA.
BOILER TEST — SHORT WORK FORM — SHEET No. 1
Test No. ____ Run No. ____ Date ____ Calculated By ____

	18th. May 1920	19th. May 1920	19th. May 1920	20th. May 1920	19th. May 1920	20th. May 1920	20th. May 1920	21st. May 1920	21st. May 1920
ITEM	Run #54	Run #55	Run # 57	Run #58	Run #59	Run #61	Run #61 a	Run #62	Run #63
ting Surface, Sq. Ft.	6065	6065	6065	6065	6065	6065	6065	6065	6065
	Oxweld	Oxweld	Oxweld	Oxweld	Oxweld	Oxweld	Oxweld	Oxweld	Oxweld
e	B & W	B & W	B & W	B & W	B & W	B & W	B & W	B & W	B & W
ttes, Tips & Plugs	6065	6065	6065	6065	6065	6065	6065	6065	6065	6065
es	4	4	4	4	4	4	4	4	4	4
Burners	2-2-2-2	1-2-2-2	2-2 2-2	2-2-2-2	2-2-2-2	2-2-2-2	2-2-2-2	2-2-2-2	2-2-2-2	2-2-2-2
Fuel, BTU	18840	18840	18840	18840	18840	18840	18840	18840	18840	18840
(CO2) %	12.8	11.8	12.6	12.9	13.7	12.2	11.7	12.4	12.4	12.4
	5.8	5.8	5.7	5.4	5.7	4.4	4.8	4.8	4.8	4.1
(CO) %	0	0	0	0	0	0	0	0	0	0
	66.7	66.2	66.5	66.7	66.6	66.4	66.5	66.4	66.5	66.5
ble Temp—Lbs. Sq. In.	434.9	434.7	434.9	434.9	434.9	434.6	434.9	434.9	434.9	434.9
al Mercury Turbine Temp—Lbs. Sq. In. Gauge										
meters, Ins. of Mercury	30.32	30.30	30.32	30.31	30.34	30.34	30.33	30.30	30.34	30.34
e Beam, Ins. of Water	2.32	2.42	2.19	2.33	2.32	2.13	2.32	2.31	2.31	2.27
ning of Smoke Pipe	-.16	-.16	-.17	-.17	-.16	-.20	-.20	-.20	-.20	-.17
ion, Ins. of Water	0.32	0.32	0.26	0.31	0.29	0.32	0.30	0.32	0.32	0.32
e S.H. Inlet B 'F	READINGS TAKEN AT IRREGULAR INTERVALS - Steam practically 100% quality.									
Boiler Outlet 'F ng. of Right & Left Leg)	667	675	664	667	697	668	673	697	668	
on Pass 'F	223.6	222.7	210.9	220.6	223.6	214.8	216.7	223.7	220.6	
ed Water at Boiler 'F	217.9	220.8	221.1	218.5	218.0	217.1	217.1	217.2	217.5	
y Duct, 'F	102.9	102.9	114.6	104.9	98.1	102.9	104.7	79.8	110.5	
t Duct, 'F	73.6	73.6	77.8	73.2	73.0	73.3	74.7	102.9	74.2	
Meter	2.0	2.0	2.0	1.8	1.0	2.7	1.8	1.35	2.0	
	31147	4838	4834	3787	3821	7817	3888	3141.5	3839	
perature of S.W.H. Superheater—Ins.	68860	68166	68488	68088	68088	166976	68488	48884	67868	
Meter—Lbs	3871.2	3848	3827	3831.2	3821	3831.6	3839.2	3812.2	3836.0	
Water Per Sq. Ft. Boiler ng Surface Lbs	0.6336	0.633	0.627	0.630	0.630	0.630	0.627	0.628	0.632	
ervac Per Hour	644	632	804.0	632.0	630 ✓	800	807.2	630	640	
ppeared by Water ter & Superheater Ft.	24885	27804	27714	25972	25180	27880.0	27885.0	28847.2	25780.0	
ctors for Boiler water ter & Superheater	1.1982	1.1986	1.1981	1.1970	1.1988	1.1982	1.1913	1.1969	1.1983	
ration of Boiler Water ter Heating Surface	41727.3	38815.9	38788.4	40468.7	40448.9	38484.2	38818.1	40888.4	40438.7	
ration of Boiler Water ter Heating Surface Per Pound Coal	16.887	16.022	16.176	16.075	16.128	15.978	16.026	16.126	16.074	
Per Lb of Oil—BTU	18764.7	18847.7	18827.2	18879.8	18844.0	18808.1	18840.7	18867.8	18880.2	
ency of Boiler Water Heating Surface Including Superheater %	82.80	82.85	82.82	82.80	82.84	82.88	82.84	82.16	82.79	

FIGURE 18.

Abridged Data Sheet Showing Pertinent Data During Certain Side Firing Runs At 300# - 450# And 600# Working Pressures.

		900#				450#				600#				
1	Run Number	84	72	68	61	65	69	76	78	66	73	75	79	63
13	Pressure Pounds Value - Gr.Ft.	485.6	485.6	485.6	485.6	485.6	485.6	485.6	485.6	485.6	485.6	485.6	485.6	485.6
16	Size of Orifice Plates	4000	3000	2000	2000	4000	3000	2000	2000	4000	3000	2000	2000	2000
18	Number of Nozzles	3	4	5	6	3	4	5	6	3	4	4	5	6
23	Arrangement of Nozzles	4-5-6	3-5-5-6	3-3-4-5-6	All	4-5-6	3-5-5-6	3-3-4-5-6	All	4-5-6	3-5-5-6	3-3-4-5-6	3-3-4-5-6	All
25	B.T.U. Number of Oil	8-1885	8-1885	8-1885	8-1885	8-1885	8-1885	8-1885	8-1885	8-1885	8-1885	8-1885	8-1885	8-1885
26	Carbon - Percent	86.48	86.99	86.65	86.48	86.99	86.99	86.52	86.52	86.48	86.48	86.48	86.48	86.48
27	Hydrogen - Percent	11.03	12.37	11.03	11.03	11.48	11.48	11.44	11.44	11.03	11.03	11.03	11.03	11.03
28	Sulphur - Percent	1.95	1.64	1.95	1.95	1.73	1.73	1.94	1.94	1.95	1.95	1.95	1.95	1.95
29	Nitrogen - Percent	0.39	0.39	0.39	0.39	0.22	0.22	0.24	0.24	0.39	0.39	0.39	0.39	0.39
30	Oxygen and Undetermined - Percent	0.31	0.77	0.31	0.31	0.42	0.42	0.16	0.16	0.31	0.31	0.31	0.31	0.31
35	Free Moisture in Oil	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace
36	Calorific Value B.T.U.	18635	18900	18635	18635	18681	18673	18673	18732	18635	18635	18635	18635	18635
37	Carbon Boiling - Percent CO2	10.9	12.2	12.2	12.5	12.1	12.7	12.2	11.8	11.1	12.1	12.8	12.4	12.7
38	Oxygen - Percent O2	4.2	4.4	3.0	4.1	4.8	3.7	3.0	4.9	4.4	3.3	4.4	4.1	3.7
39	Carbon Monoxide - Percent CO	0	0	0	0	0	0	0	0	0	0	0	0	0
40	Nitrogen - Percent N2	83.8	83.4	83.7	83.4	83.4	83.6	84.8	83.3	82.9	82.7	82.8	83.5	83.4
42	Press. of Feed - Lbs. per Sq. In. Abs.	209.3*	216.4	216.1	208.1	465.7	441.1	449.4	449.3	612.2*	610.7	602.4	612.2	615.1
51	Press. Outlet Superheater - Lbs. per Sq. In. Abs.	201.6	211.1	206.9	208.3	461.7	438.8	446.1	447.4	606.7	607.6	598.9	609.9	608.4
52	Press. Fireside - Lbs. of Water	1.27	2.20	2.06	0.36	1.14	2.61	2.56	0.84	1.47	2.41	2.74	2.68	0.44
53	Press. at Regulator - Lbs. of Water	-0.10	-0.19	-0.16	-0.16	-0.16	-0.16	-0.20	-0.20	-0.1	-0.20	-0.19	-0.22	-0.16
54	Press. of Furnace - Lbs. of Water	0.20	0.79	2.00	2.07	0.25	0.62	1.80	2.0	0.16	0.62	0.72	1.80	2.0
70	Temp. of Steam at Superheater Outlet - Average °F.	665	665	679	691	621	683	700	716	680	695	700	720	720
72	Temp. Saturation Steam at Main Press. - °F.	480.8	481.7	482.5	482.8	489.6	488.8	489.4	490.8	488.3	488.1	487.9	488.3	488.8
73	Superheat Based on Main Press. - °F.	181.7	181.3	196.7	208.7	131.6	195.2	210.1	226.2	191.7	206.9	212.8	231.7	231.2
77	Superheat Based on Press. at Outlet - °F.	184.8	182.9	198.6	215.6	138.6	206.6	246.3	227.8	142.6	207.8	213.6	232.1	242.7
78	Temp. Feed-Water at Feed Stop - °F.	211.2	210	214.8	211.1	210.1	210.9	211.2	210.9	212.8	210.2	210.2	209.8	211.6
80	Temp. of Gases Leaving Boiler - °F.	414.6	448.6	472.8	510.0	426.2	469.6	504.0	539.0	462.8	506.4	506.0	538.8	568.2
81	Temp. of Hot Gas - °F.	180.7	118.9	118.9	121.6	111.6	100.6	118.3	126.8	128.1	114.8	1 21.1	118.6	128.2
82	Temp. of Hot Gas - °F.	91.9	88.9	93.1	89.8	86.9	89.7	90.9	94.3	94.4	92.7	91.7	88.8	94.0
100	Oil Burned Per Hour - Lbs.	1289	2044.1	2685.8	3161.8	1241	2044	2783.3	3270	1245	2245	2889	3227	3267.9
101	Oil Burned Per Hour Per Sq. Ft. of Heating B.T.U.S. - Lbs.	0.368	0.501	0.722	0.998	0.366	0.500	0.746	0.927	0.366	0.551	0.868	0.725	0.927
102	Oil Burned Per Hour Per Sq. Ft. of B.T.U.S. Approx. 19,000 BTU Oil - Lbs.	0.345	0.496	0.727	0.972	0.360	0.491	0.735	0.906	0.365	0.491	0.853	0.740	0.977
103	Oil Burned Per Hour Per Sq. Ft. of Furnace Volume - Lbs.	2.97	4.61	9.69	11.9	2.98	4.61	8.96	11.97	2.98	5.01	6.64	9.05	11.96
104	Oil Burned Per Hour Per Burner - Lbs.	429.6	681.0	764.6	841.9	413.8	681.0	788.7	846.0	421.0	636.2	659.0	707.4	846.6
105	Actual Water Evaporated Pr B.H.P. and B.T.U. Per Hour - Lbs.	17280.6	24288.7	29988.0	28868.5	18282.5	28488.0	29420.0	28139.7	16934.0	28854.0	28946.0	28888.0	28885.7

106	OIL BURNED PER HOUR PER SQ. FT. OF FURNACE VOLUME - LBS.	5.97	6.01	9.65	11.9	5.98	6.01	8.96	11.97	5.96	6.01	8.96	9.86	11.96
106	OIL BURNED PER HOUR PER BURNER - LBS.	419.6	626.0	701.6	841.9	615.3	626.0	702.7	842.9	681.0	626.3	699.0	707.4	844.6
108	ANNUAL HEAT REQUIRED BY B.T.U.S. AND B.T.U. PER HOUR - LBS.	17800.6	26886.7	29999.0	44966.3	18899.5	26485.0	40400.0	48659.7	18984.0	26854.0	39966.0	48883.0	64883.7
107	QUALITY OF STEAM AT DESIRED INLET - PERCENT	99.99	99.94	99.93	99.87	99.89	99.94	99.91	99.85	99.89	99.86	99.83	99.83	99.84
110	RATIO OF EVAPORATION PER B.T.U.S. AND B.T.U.	1.1694	1.2848	1.2876	1.2821	1.1690	1.2889	1.2186	1.2884	1.1789	1.2846	1.2877	1.2886	1.2849
110a	RATIO OF EVAPORATION PER B.T.U.S. ACTUAL	1.0889	1.0966	1.0894	1.0886	1.0849	1.0966	1.0896	1.0889	1.0894	1.0911	1.0922	1.0919	1.089
111	EQUIVALENT EVAPORATION PER B.T.U.S. AND B.T.U. PER HOUR - LBS.	20112.7	41898.6	61464.8	89799.4	19664.1	40899.1	66186.8	89458.7	19461.9	40885.7	60181.5	89881.8	79879.8
112	EQUIVALENT EVAPORATION OF B.T.U.S. AND B.T.U. PER LB. OF OIL AS FIRED - Lbs.	16.976	16.297	14.977	15.982	16.846	16.864	16.979	16.689	16.726	16.709	16.671	16.721	16.668
113	EQUIVALENT EVAPORATION OF B.T.U.S. AND B.T.U. PER LB. OF OIL ABSORBED 15000 B.T.U.	16.297	16.419	16.481	16.294	16.116	16.148	16.168	16.994	16.948	16.947	16.897	16.869	16.990
113a	EQUIVALENT EVAPORATION OF B.T.U.S. ACTUAL PER LB. OF OIL AS FIRED	16.297	16.253	16.271	16.765	16.296	16.870	16.981	16.681	16.699	16.708	16.689	16.277	16.445

		300"				450"				600"				
		66	70	80	81	66	69	74	76	66	70	76	78	86
114	BOILER HORSEPOWER	888.0	1282.6	1702.6	2550.6	878.0	1269.0	1701.0	2554.6	876.7	1269.4	1705.0	1701.0	2566.7
115	FURNACE RATING B.T.U.S. PER HOUR PER SQ. FT.	304.30	318.00	326.30	419.0	301.00	318.00	311.00	414.00	300.00	307.00	307.0	312.00	411.00
117	EQUIVALENT EVAPORATION PER HOUR PER SQ. FT. OF FURNACE B.T.U.S. AND B.T.U. - Lbs.	5.99	7.00	10.000	14.000	5.91	7.00	10.75	14.000	5.95	7.15	7.17	10.704	14.196
118	FURNACE HEIGHT AND OVER TEMPERATURE AND. RESPECT TO FURNACE CONDITIONS	45.76	37.36	29.00	26.00	35.97	28.00	26.00	20.07	41.00	29.00	22.00	27.00	24.00
120	HEAT LOSS; LOSSES DUE TO IN PER. HEAT LOSS, OIL AS BOILER TO EXHAUST, % PER LOSS, AT BOIL PRESS. - B.T.U. (DRY-GAS BASIS)	6.06 1181.1	6.08 1288.6	6.10 1184.1	6.07 1180.9	6.04 1819.6	6.04 1819.8	6.09 1184.4	6.07 1196.9	6.08 1168.8	6.12 1188.6	6.04 1188.8	6.09 1169.4	6.09 1181.1
127	HEAT LOSS; GAS TEMPERATURE IN PER. PERS LOSS, OIL AS BOILER TO EXHAUST TEMP. AT BOIL PRESS.- B.T.U. AND PERCENT	0	0	0	0	0	0	0	0	0	0	0	0	0
128	HEAT LOSS; BURNING IN UNBURNED GAS (B.T.U. AND % PERCENT LOSS, TO EXHAUST TEMP. AT BOIL PRESS.- PER CENT	0.35 65.7	0.35 67.0	0.39 66.8	0.39 68.6	0.38 68.6	0.16 74.0	0.39 66.8	0.26 67.4	0.36 66.8	0.36 66.6	0.31 67.1	0.30 66.2	0.32 69.1
129	HEAT LOSS DUE UNBURNED CARBON (B.T.U. LOSS DUE UNBURNED CARB. TO EXHAUSTED TEMP. AT BOIL PRESS - B.T.U. & PERCENT	5.16 966.0	5.46 2047.1	5.46 1618.6	5.4 1606.6	6.39 1178.4	6.39 1176.3	6.17 1188.9	6.06 1128.3	6.46 1207.3	6.66 1260.0	6.49 1210.6	6.74 1268.6	6.36 1188.3
130	TOTAL UNACCOUNTABLE LOSSES - B.T.U. AND (% PERCENT	11.47 2139.6	12.10 2266.2	11.63 2207.1	11.7 2181.0	13.69 2562.6	13.64 2438.3	12.66 2366.6	12.75 2397.2	13.06 2438.0	13.30 2469.0	13.04 2438.0	13.36 2479.1	12.9 2405.9
131	HEAT LOSS; RADIATION OF CARBON - B.T.U. AND PERCENT.	0	0	0	0	0	0	0	0	0	0	0	0	0
132	HEAT LOSS; SENSIBLE AIR REQUIRED PERCENT (B.T.U. AND PERCENT. AIR, BOIL PRESS PERCENT LOSS, TO EXHAUST. AT BOIL PRESS.	1.30 284.2	1.37 294.6	1.39 294.7	1.47 295.4	1.35 248.1	1.47 276.9	1.36 257.7	1.61 299.7	1.47 488.7	1.47 272.0	1.93 288.1	1.93 288.1	1.65 286.8
133	HEAT LOSS; LOSS-HEAT CONT. OF ASH IN FUEL HEAT CONT. OF ASH IN BOILER AT BOIL PRESS. TO GAS-BASIS LOSSES (B.T.U. BOILER	0 0	0.06 21.0	0.10 28.8	0.21 30.7	0 0	0.06 9.9	0.11 29.7	0.19 36.3	0 0	0.06 0.3	0.04 0.3	0.12 22.8	0.18 33.6
140	HEAT LOSS; HEAT RADIATION FUEL HEAT CONT. AT BOIL PRESS. TO CONT. OF GASES LEAVING BOILER - B.T.U. AND PERCENT	0	0	0	0	0	0	0	0	0	0	0	0	0
145	HEAT LOSS; MISC. IN UNACCOUNTED FOR (%	0	0.02	0.06	0.07	0	0.01	0.03	0.07	0	0.02	0.02	0.04	0.07

100	OIL BURNED PER HOUR - LBS.	1820	3644.1	5035.2	8051.6	1861	3846	5793.3	8079	1863	3845	5869	8057	8067.9
101	OIL BURNED PER HOUR PER SQ. FT. OF BOILER H.S.S. - LBS.	0.368	0.921	0.722	0.995	0.366	0.930	0.746	0.997	0.368	0.921	0.863	0.756	0.997
102	OIL BURNED PER HOUR PER SQ. FT. OF H.S.S. ASSUMING 19,000 BTU OIL - LBS.	0.363	0.496	0.727	0.975	0.360	0.492	0.733	0.983	0.363	0.491	0.693	0.740	0.977
103	OIL BURNED PER HOUR PER CU. FT. OF FURNACE VOLUME - LBS.	2.97	6.01	9.03	11.9	2.93	6.01	8.95	11.97	2.96	6.01	6.04	9.06	11.96
104	OIL BURNED PER HOUR PER HORSE - LBS.	619.6	636.0	764.6	861.9	615.8	636.0	788.7	868.0	621.0	636.2	699.8	767.4	866.6
105	ACTUAL RATE EVAPORATED BY H.S.S. AND S.H.S. PER HOUR - LBS.	17280.0	34286.7	48900.0	68066.3	16888.8	36488.0	49400.0	68638.7	16934.0	35256.0	53866.0	69286.0	84868.7
107	QUALITY OF STEAM AT SUPERHEAT INLET - PERCENT	99.90	99.84	99.92	99.87	99.89	99.84	99.81	99.85	99.80	99.86	99.68	99.88	99.84
110	RATIOS OF EVAPORATION FOR H.S.S. AND S.H.S.	1.1694	1.2548	1.2976	1.2822	1.1696	1.2629	1.2186	1.2884	1.1720	1.2846	1.2877	1.2888	1.2869
110a	RATIOS OF EVAPORATION FOR H.S.S. ALONE	1.0888	1.0966	1.0894	1.0825	1.0847	1.0858	1.0886	1.0889	1.0896	1.0911	1.0881	1.0819	1.080
111	EQUIVALENT EVAPORATION FOR H.S.S. AND S.H.S. PER HOUR - LBS.	20113.7	41288.6	61466.8	86789.6	19686.1	42289.1	60198.8	88455.7	19681.9	40588.7	46188.8	80686.6	79478.8
112	EQUIVALENT EVAPORATION OF H.S.S. AND S.H.S. PER LB. OF OIL AS FIRED - LBS.	16.976	16.197	16.877	16.988	16.846	16.866	15.870	15.886	16.726	16.736	16.691	16.762	16.682
113	EQUIVALENT EVAPORATION OF H.S.S. AND S.H.S. PER LB. OF OIL ASSUMING 19000 B.T.U.	16.297	16.410	16.682	16.986	16.116	16.162	16.168	16.054	16.843	16.867	16.827	16.660	16.996
113a	EQUIVALENT EVAPORATION OF H.S.S. ALONE PER LB. OF OIL AS FIRED	16.297	16.313	16.972	16.768	16.288	16.070	16.781	16.681	16.678	16.726	16.669	16.577	16.665

		300#				450#				600#				
		66	70	80	81	68	69	74	75	65	79	76	78	82
114	BOILER EVAPORATION	8888.0	2282.6	2782.6	3888.6	878.0	1180.0	1764.0	2882.6	878.2	1180.4	2288.0	2781.0	2886.7
116	FURNACE LOSSES H.S.S.-S.H.S. INCL. 1 CU	104.18	218.88	286.88	416.0	101.88	206.88	211.64	414.68	100.68	207.88	207.0	218.88	411.48
117	EQUIVALENT EVAPORATION PER HOUR PER SQ. FT. OF HEATING SURFACE H.S.S. AND S.H.S. - LBS.	8.88	7.88	10.778	14.488	8.81	7.11	10.78	14.888	8.88	7.18	7.17	10.784	14.198
118	PERCENT HEATING AIR OVER THEORETICAL AMT. REQUIRED TO COMPLETE COMBUSTION	48.78	87.81	18.48	88.88	81.87	88.88	18.88	88.87	61.88	88.88	88.88	87.48	84.88
120	HEAT LOSS, HEATING GAS TO 1st INCL. FURN. TEMP., CU. AT BOILER TO SATURATED STEAM TEMP. AT BOIL PRESS. - B.T.U. (PER- CENT BOILER)	6.86 1181.1	6.88 1888.8	6.18 1184.1	6.87 1184.8	6.84 1828.6	6.86 1818.8	6.88 1184.4	6.87 1184.8	6.88 1168.8	6.18 1168.4	6.86 1168.8	6.88 1148.4	6.88 1141.1
127	HEAT LOSS, GAS LEAVING 1st INCL. FURN. TEMP., CU. AT BOILER TO SATURATED STEAM TEMP. AT BOIL PRESS.- B.T.U. AND PERCENT	0	0	0	0	0	0	0	0	0	0	0	0	0
128	HEAT LOSS, HEATING AIR TEMPER-AIR (# FURN TEMP. TO SATURATED STEAM AT BOIL PRESS.-BTU AND PERCENT (PER	0.88 48.7	0.88 47.0	0.89 86.8	0.89 48.6	0.88 48.6	0.18 86.6	0.88 86.8	0.82 88.6	0.88 88.2	0.88 88.6	0.81 87.1	0.80 86.2	0.82 88.1
129	HEAT LOSS WITH UNBURNT DRY GAS. PIER (# GASES FROM FURNACE TEMP. TO SATURATION TEMP. AT BOIL PRESS - B.T.U. & PERCENT (PER	6.18 988.8	6.48 1887.1	6.48 1818.8	6.4 1886.8	6.88 1178.4	6.88 1178.8	6.18 1182.9	6.88 1182.8	6.48 1897.8	6.88 1880.8	6.48 1818.4	6.69 1888.8	6.84 1186.8
130	HEAT UNAVOIDABLE LOSSES - B.T.U. AND (# PERCENT (PER	11.47 3180.8	12.18 2878.8	11.81 2887.1	11.7 2181.8	12.88 2868.4	12.88 2888.8	12.84 2886.4	12.78 2887.8	12.84 2828.8	12.88 2869.8	12.84 2888.8	12.84 2878.1	12.9 2868.8
141	HEAT LOSS, HEATED CONSTITUENTS OF GASES - B.T.U. AND PERCENT.	0	0	0	0	0	0	0	0	0	0	0	0	0
146	HEAT LOSS, HEATED AIR LEAVING BOILER (# AND DEGS. ABS. ABOVE FINAL FURNACE TEMP. TO SAT. TEMP. AT BOIL PRESS. (PER	2.88 484.2	1.87 294.6	1.88 284.7	1.47 278.4	1.88 848.1	1.47 278.9	1.86 287.7	1.61 288.7	8.47 488.7	1.47 272.8	1.98 288.1	1.98 288.1	1.65 285.8
148	HEAT LOSS: RADIATION OF HEAT IN (# FUEL, GAS TEMP. OF ENT IN BOILER AT BOIL PRESS. TO TEMP ALONE LEAVING BOILER (PER	0 0	0.86 11.8	0.18 88.8	0.81 88.7	0 0	0.88 8.9	0.11 88.7	0.18 86.8	0 0	0.88 8.8	0.84 8.8	0.12 88.8	0.18 88.6
144	HEAT LOSS: HEAT RETAINED FUEL, GAS &C. TEMP. AT BOIL PRESS. TO TEMP. OF GASES LEAVING BOILER - B.T.U. AND PERCENT	0	0	0	0	0	0	0	0	0	0	0	0	0

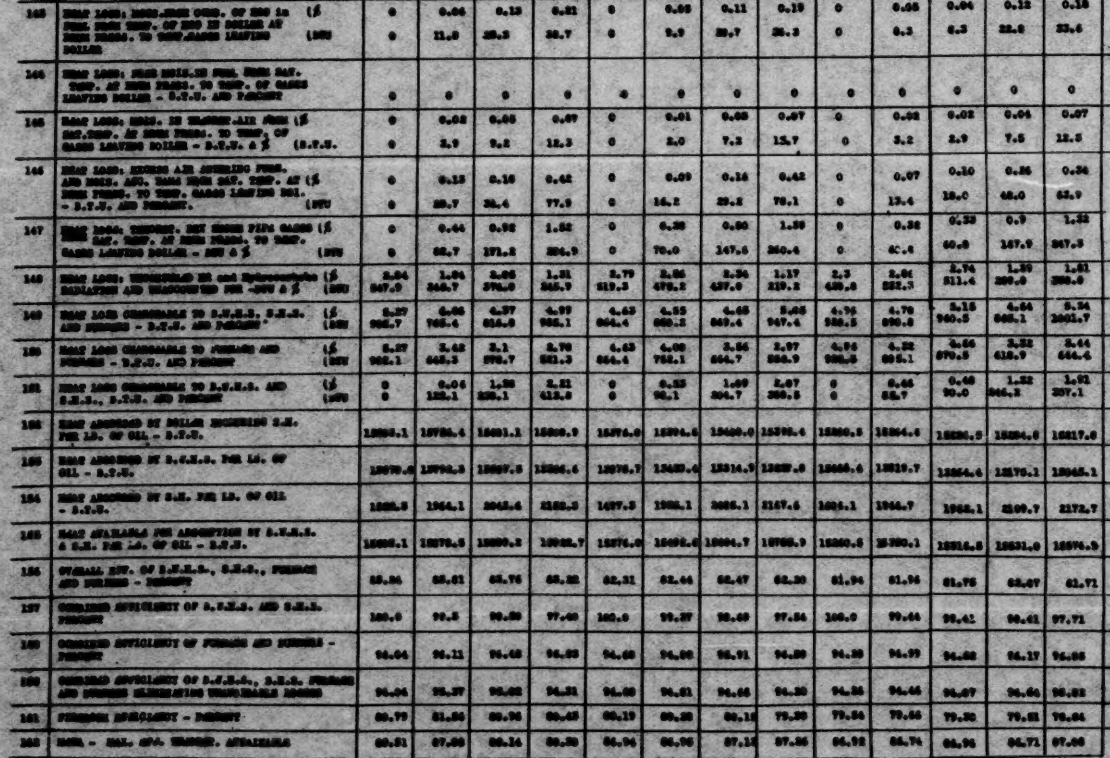


FIGURE 20.

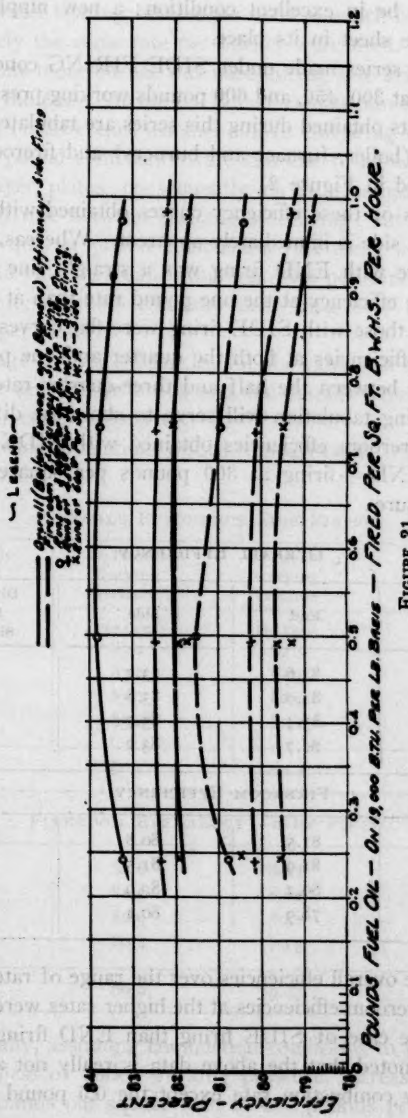

Efficiency - Percent
Pounds Fuel Oil — On 19,000 B.T.U. Per Lb. Basis — Fired Per Sq. Ft. B.H.S. Per Hour.

FIGURE 21.

and found to be in excellent condition; a new nipple was then rolled into the sheet in its place.

The second series made under SIDE FIRING conditions consisted of runs at 300, 450, and 600 pounds working pressures. The principal results obtained during this series are tabulated in Figure 20. Overall (boiler, furnace and burners) and fireroom efficiencies are plotted in Figure 21.

The flatness of these efficiency curves obtained with the boiler fired from the side is immediately apparent. Whereas, the overall efficiency curve with END firing was a straight line with about 2 per cent less efficiency at the one pound rate than at the quarter (Figure 12); those with SIDE firing were flat curves with practically equal efficiencies at both the quarter and one pound rates, maxima being between the half and three-quarter rates.

The following tabulation will serve to show the differences in overall and fireroom efficiencies obtained with SIDE — as compared with END — firing at 300 pounds per square inch gage working pressure.

OVERALL EFFICIENCY.

Approximate Combined Rate	End	Side	Difference in Favor of Side Firing
1/4	84.6	83.2	—0.6
1/2	84.0	83.8	—0.2
3/4	83.4	83.7	+0.3
1	82.7	83.2	+0.5
	FIREROOM EFFICIENCY.		
1/4	81.5	80.8	—0.7
1/2	81.9	81.5	—0.4
3/4	80.2	80.9	+0.7
1	78.9	80.4	+1.5

The average overall efficiencies over the range of rates run were the same. Fireroom efficiencies at the higher rates were materially superior in the case of SIDE firing than END firing, however.

It is to be noted that the above data is really not a just comparison at any combustion rate except the 0.5 pound per square

foot boiler water heating surface per hour at which four burners at approximately the same rate per burner were used. At the other three rates one more burner was used during side firing than when firing from the end. At the 0.5 pound combustion rate there was substantially no difference whether side or end firing was employed. No quarter pound combustion rate run end firing was made with 3-4308 sprayer plates, consequently no true comparison can be made at that rate between end and side firing; but in other runs at the three-quarter pound combustion rate using 5-3208 sprayer plates overall efficiency end-firing was 83.3 per cent — side firing 83.7 per cent; fireroom efficiency end firing 80.6 per cent, side firing 80.9 per cent.

The average overall efficiencies for the rates run at 300 pounds was 83.5 per cent; those at 450 and 600 pounds were 82.4 per cent and 81.9 per cent, respectively.

The following tabulations afford comparison of overall and fireroom efficiencies:

OVERALL EFFICIENCY SIDE FIRING.

Approximate Combined Rate	300 Pounds Working Pressure	450 Pounds Working Pressure	600 Pounds Working Pressure
	Per Cent	Per Cent	Per Cent
1/4	83.2	82.3	81.9
1/2	83.8	82.5	82.0
3/4	83.7	82.5	82.1
1	83.2	82.2	81.7
Average:	83.5	82.4	81.9

FIRROOM EFFICIENCY — SIDE FIRING.

1/4	80.8	80.2	79.6
1/2	81.5	80.2	79.7
3/4	80.9	80.2	79.5
1	80.4	79.3	78.8
Average:	80.9	79.98	79.4

Theoretically, assuming combustion conditions to be exactly the same regardless of boiler working pressure, increase in pressure from 300 pounds per square inch to 450 pounds per square inch

would cause a lowering of 0.9 per cent in overall efficiency, advance in pressure to 600 pounds per square inch would entail a further loss of 0.7 per cent, or — for the total increment from 300 to 600 pounds per square inch — loss would be 1.6 per cent.

The method of calculating these reductions in efficiency was as follows:

Assuming 18.5 pounds of gas produced per pound of fuel oil, the latter having a calorific value of 19,000 B.T.U.

300 pounds to 450 pounds W. P. 456.3–417.3 = 39

$$\frac{18.5 \times .24 \times 39}{19{,}000} = 0.91 \text{ per cent}$$

450 pounds to 600 pounds W. P. 486.3–456.3 = 30

$$\frac{18.5 \times .24 \times 30}{19{,}000} = 0.7$$

300 pounds to 600 pounds W. P. = 1.61 per cent.

The fireroom air pressures required for trace smoke (shown in Figure 22) ranged from approximately 1.5 inch (of water) at the quarter pound combustion rate to slightly over eight at the one pound rate. While not markedly lower than with END firing at the quarter and half pound rates, they were considerably less at the 0.75 pound and 1.0 pound rates; minimum fireroom air pressures required at the latter two rates end firing were 6.4 and 11.6 inch respectively, under side firing conditions these were approximately 5.4 and 8.2 inch. This was principally due to use of a larger number of burners, 5 instead of 4 being used at the 3/4 pound rate, 6 instead of 5 at the 1 pound rate.

The furnace pressures under side firing conditions ranged from approximately 0.2 inch at the 0.25 pound combustion rate to about 3.0 inch at the 1.0 pound combustion rate, being slightly higher throughout the entire range than when end firing. The furnace was oblong and when fired from the side only 6 feet was available as compared with more than 10 feet when fired from the end.

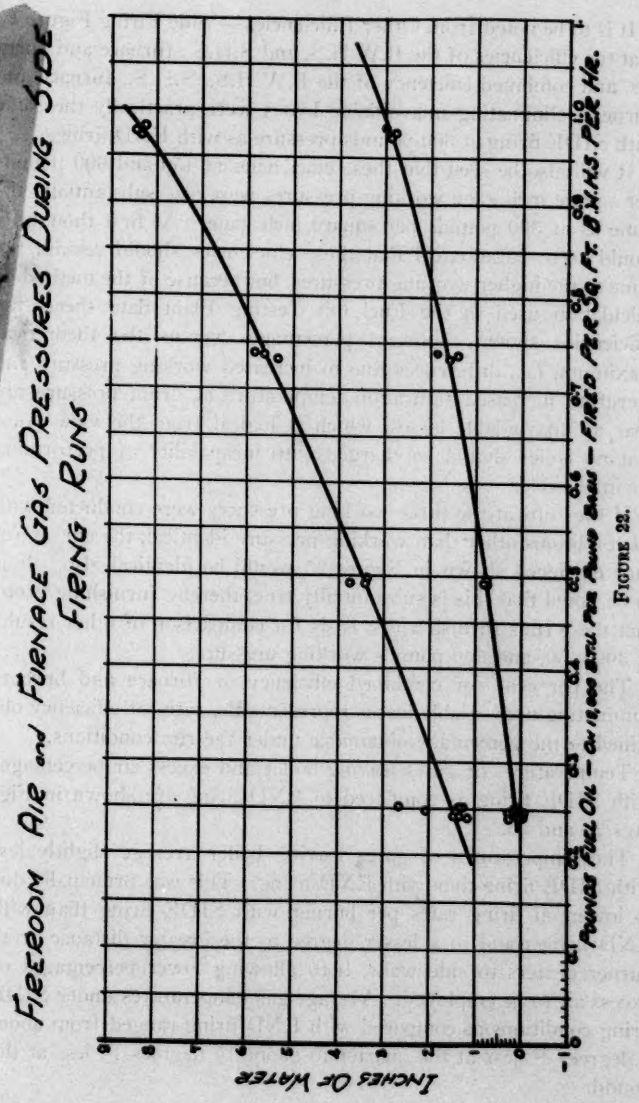

Fireroom Air and Furnace Gas Pressures During Side Firing Runs
Inches of Water
Pounds Fuel Oil — on 19,000 B.T.U. per Pound Basis — Fired per Sq. Ft. B.H.S. per Hr.

FIGURE 22.

It is to be noted from Other Efficiencies — Side Firing Figure 23 that the efficiencies of the B.W.H.S. and S.H.S., furnace and burners, and combined efficiency of the B.W.H.S., S.H.S., furnace and burners, eliminating unavoidable losses were practically the same with SIDE firing at 300 pounds pressure as with END firing.

It will also be seen that these efficiencies at 450 and 600 pounds per square inch gage working pressures were also substantially the same as at 300 pounds per square inch gage. At first thought it would seem paradoxical that these efficiencies should remain the same at the higher working pressures, but because of the method of calculation used in the Fuel Oil Testing Plant data sheets the efficiencies shown represent percentages versus the theoretical maximum, *i.e.*, differences due to increased working pressure and therefore increased saturation temperature at drum pressure appear as unavoidable losses, which is logical from the view point that no boiler should be charged with incapability of performing the impossible.

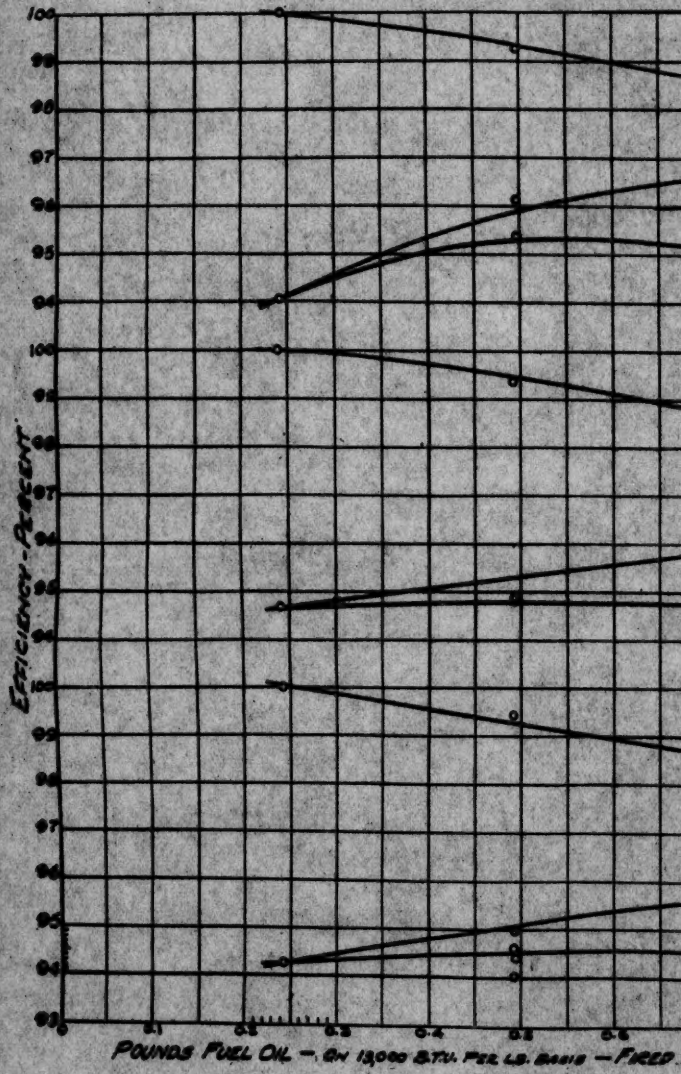
If the runs at the three working pressures were conducted with all conditions other than working pressure identical, the curves for each efficiency shown in Figure 23 would be identical also. It is to be noted that this is substantially true, thereby furnishing proof that these runs furnish a just basis for comparison of other results at 300, 450, and 600 pounds working pressure.

The per cent for combined efficiency of furnace and burners eliminating unavoidable losses represents the ratio of efficiency obtained to the percentage obtainable under the run conditions.

Temperatures of gases leaving boiler and excess air percentages with SIDE firing as compared to END firing are shown in Figures 24 and 25.

The temperatures of gases leaving boiler average slightly less with SIDE firing than with END firing. This was principally due to lower oil firing rates per burner with SIDE firing than with END firing; and to a lesser degree to the greater distance from burner centers to side walls, both allowing lower percentages of excess air to be employed. Average gas temperatures under SIDE firing conditions as compared with END firing ranged from about 4 degrees F. less at the quarter to about 15 degrees F. less at the pound.

OTHER EFFICIENCIES - SIDE



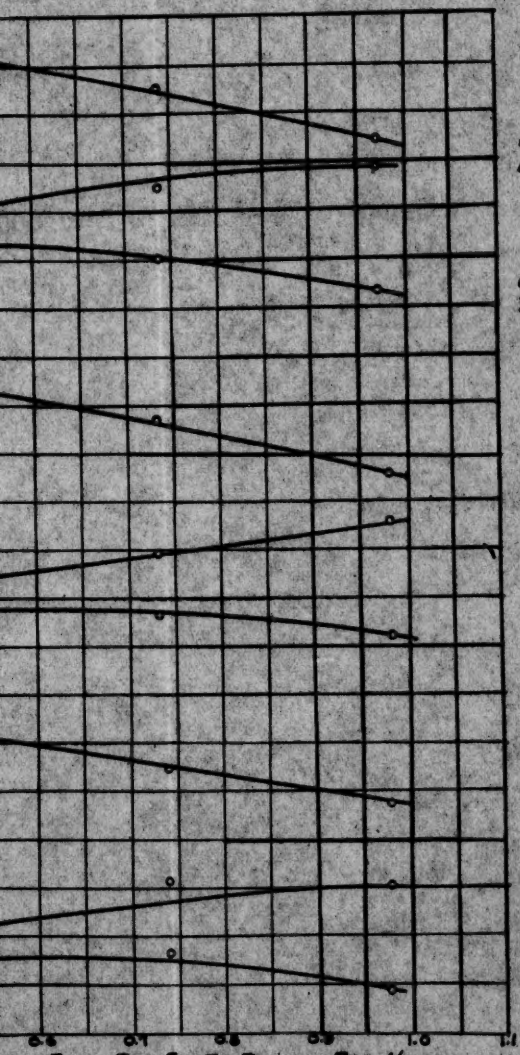

SIDE FIRING
300 #/□" WORKING PRESSURE - SIDE FIRING
Efficiency of B.W.H.S. and S.H.S.
Efficiency of Furnace and Burners
Combined Efficiency of B.W.H.S. S.H.S. Furnace and Burners Eliminating Unavoidable Losses.
450 #/□" WORKING PRESSURE - SIDE FIRING
Efficiency of B.W.H.S. and S.H.S.
Efficiency of Furnace and Burners
Combined Efficiency of B.W.H.S. S.H.S. Furnace and Burners Eliminating Unavoidable Losses.
600 #/□" WORKING PRESSURE - SIDE FIRING.
Efficiency of B.W.H.S. and S.H.S.
Efficiency of Furnace and Burners
Combined Efficiency of B.W.H.S. and S.H.S. Furnace and Burners Eliminating Unavoidable Losses
0.6
0.7
0.8
0.9
1.0
1.1
— FIRED PER SQ. FT. B.W.H.S. PER HOUR.

FIGURE 23.

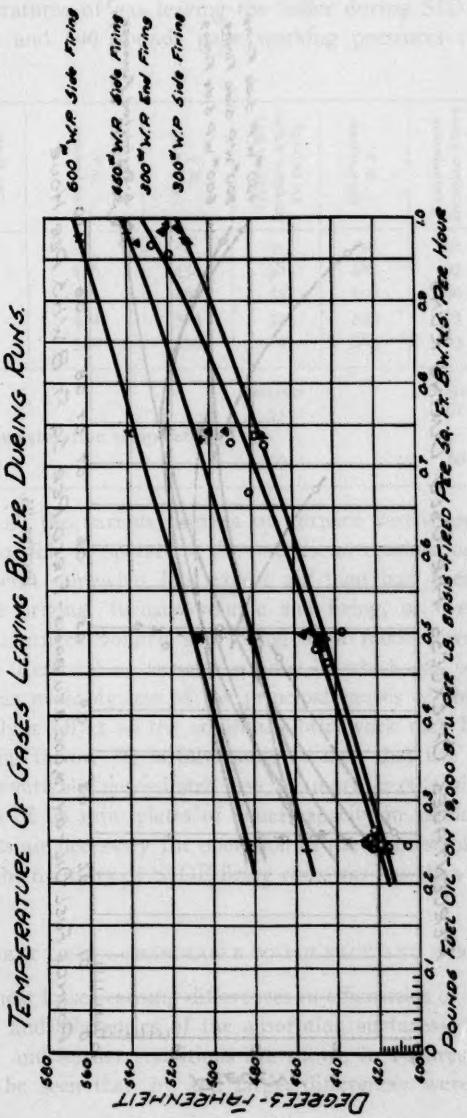

TEMPERATURE OF GASES LEAVING BOILER DURING RUNS.
600 LB. W.P. Side Firing
450 LB. W.P. Side Firing
300 LB. W.P. End Firing
300 LB. W.P. Side Firing
DEGREES - FAHRENHEIT
POUNDS FUEL OIL - OIL 18,000 BTU PER LB. BASIS - FIRED PER SQ. FT. B.H.S. PER HOUR

FIGURE 24.

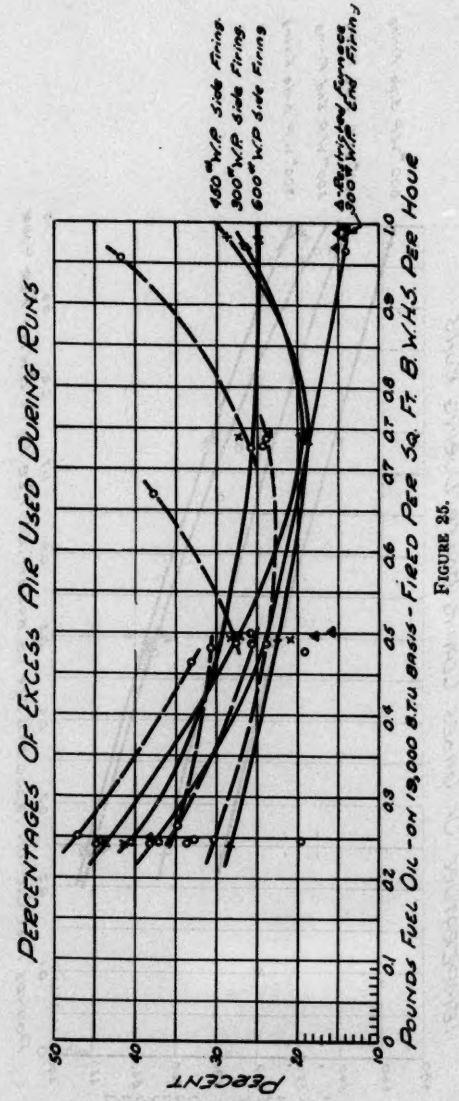

PERCENTAGES OF EXCESS AIR USED DURING RUNS
150 W.P. Side Firing
300 W.P. Side Firing
600 W.P. Side Firing
3-Furnace End Firing
PERCENT
50
40
30
20
10
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
POUNDS FUEL OIL - ON 18,000 B.T.U. BASIS - FIRED PER SQ. FT. B.W.H.S. PER HOUR

FIGURE 25.

Temperatures of gas leaving the boiler during SIDE firing at 300, 450 and 600 pounds gage working pressures compare as follows:

Rate Pounds per Square Foot B.W.H.S. per Hour	300 Pounds W.P.	450 Pounds W.P.	Temperature Increase Above 300 Pounds	600 Pounds W.P.	Temperature Increase Above 300 Pounds	Temperature Increase Above 450 Pounds
	°F.	°F.	°F.	°F.	°F.	°F.
1/4	416	454	38	483	67	29
1/2	446	481	35	506	60	25
3/4	474	504	30	539	65	35
1	510	539	29	563	63	24
			Degrees		Degrees	Degrees
Average			33		61	28.3
Increase in saturation temperature at W.P.			39		68	29.9

With all the various degrees of furnace restriction employed it was possible to operate at the one pound combustion rate trace smoke with somewhat less excess air than had been necessary with the original furnace volume end firing, or even with the original furnace volume side firing at a much lower rate per burner. Removal of furnace volume in which gas eddies might occur was probably one of the principal causes of this, although the catalytic effect of the additional brickwork may have been a qualifying factor. It is interesting to note that use of 10 ratio plates invariably necessitated use of more excess air than like numbers of 08 ratio plates of equal capacity in similar positions.

Excess air necessary for operation at the trace smoke condition during the majority of SIDE firing runs was less than while END firing.

HEAT LOSSES CHARGEABLE TO FURNACE AND BURNERS.

The heat losses causing differences in efficiencies of furnace and burners and efficiencies of the absorption surfaces—with various furnace and burner conditions are shown in Figures 26 and 27. It will be seen that, by and large, differences were principally

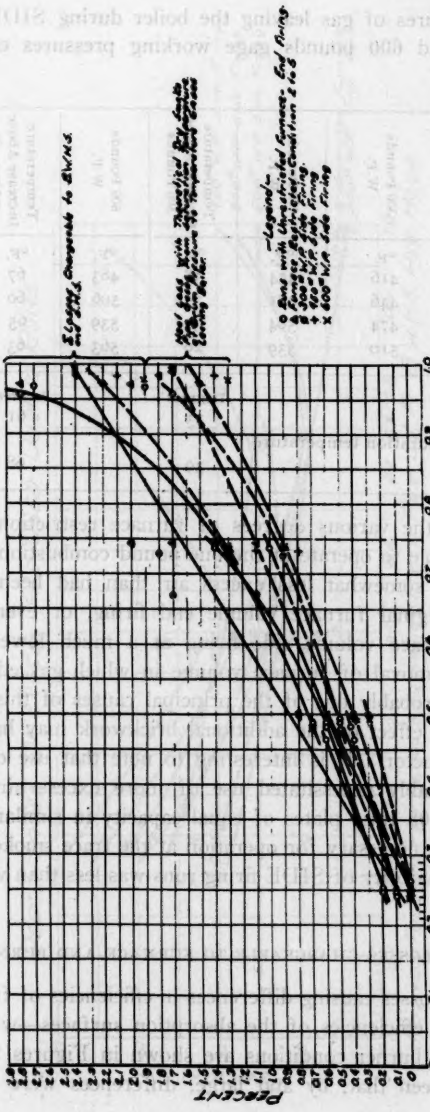
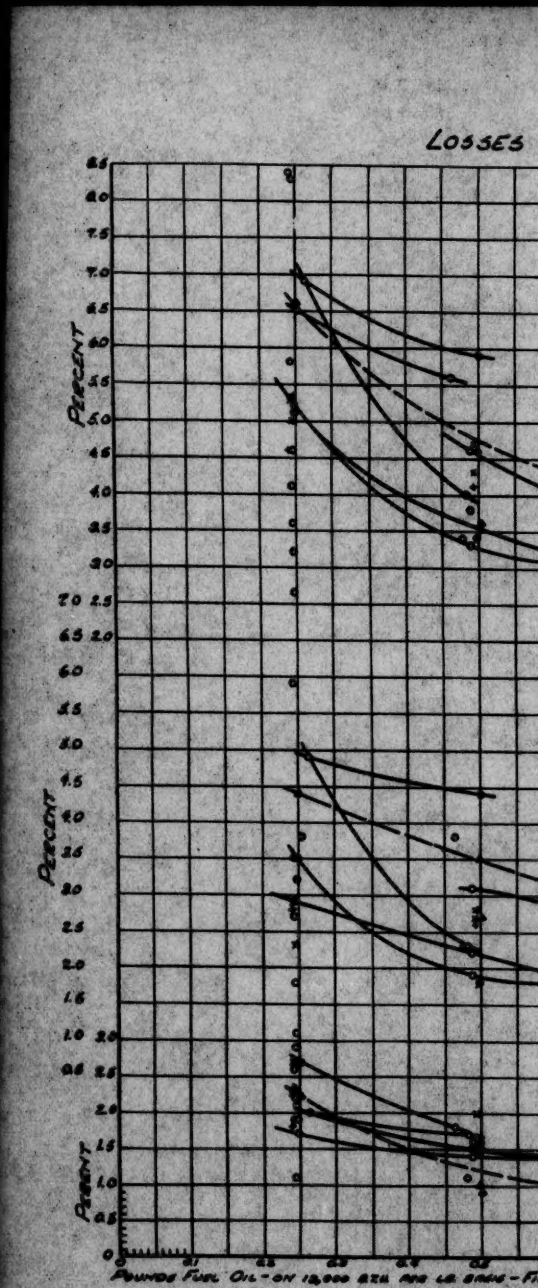

LOSSES
PERCENT
POUNDS FUEL OIL - ON 13,000 B.T.U. BASIS - FIRED PER SQ. FT. B. W. H. S. PER HOUR

FIGURE 27.


LOSSES
PERCENT
PERCENT
PERCENT
POUNDS FUEL OIL - ON 18,000 BTU AND 18 BRIX - F

LOSSES

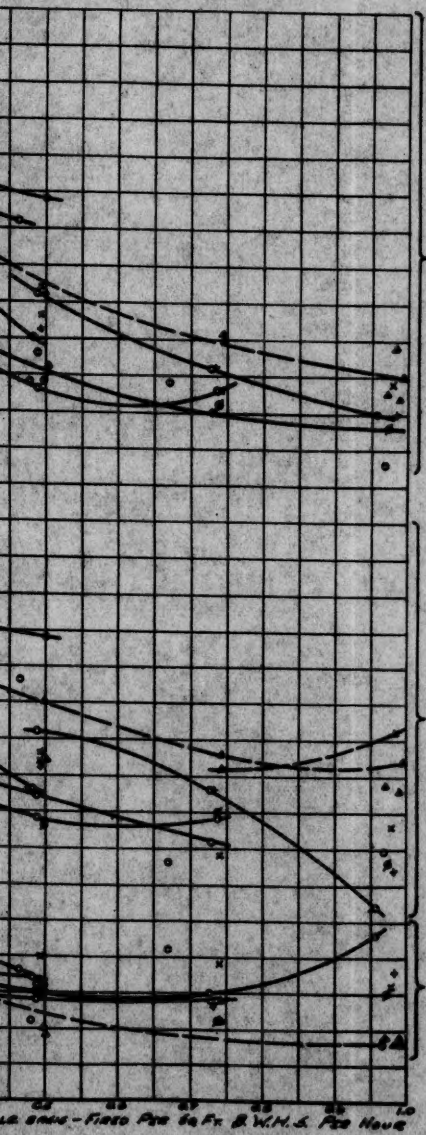

% Heat Losses Chargeable to Furnace and Burners
—Legend—
Runs with Unrestricted Furnace End Area
Runs with Restricted Furnace End Area
Side Firing
Side Firing
Side Firing
Losses due to Unburned Hydrocarbons Radiation and Unaccounted For
Losses due to Excess Air, Loss Furnace Temperature or Draft Pressure
Pounds Coke Fired Per Sq. Ft. B.T.H.S. Per Hour

FIGURE 26.

through losses due to unburned hydrocarbons, radiation and unaccounted for. Considerable study was devoted to the causes of these losses with various burner combinations. Although a complete discussion of this cannot be made in this article, one or two examples are representative.

At the quarter pound combustion rate operation with 1-2808 sprayer plate was accompanied by the highest hydro-carbon loss (over 7.1 per cent) of any condition employed, and the lowest loss due to excess air from fireroom temperature to saturation temperature at drum pressure—approximately 1.1 per cent. The high percentage of loss due to hydrocarbons under that condition is believed to have been due to the relative coarseness of the atomized oil particles with this size of sprayer plate at the rate of 1250 pounds oil per hour, and insufficient air velocity to complete combustion of gases from these particles prior to their being chilled by the tube bank. In the cases where two and three burners were used in the upper and lower positions with the same sizes of sprayer plates on accurate comparison runs hydrocarbon losses were higher and excess air lower in the case of the lower burners, but the hydrocarbon loss was greater than the saving in excess air. It would seem from this that operation of the lower burners at trace smoke required less air or less air velocity than that necessary to assure complete combustion without hydrocarbon loss, no measurable amounts of CO, however, could be detected by Orsat.

The somewhat higher furnace losses sustained with restricted furnace conditions than with the original furnace volume was apparently due to unburned hydrocarbon losses, but as previously noted, this loss is believed due to continued absorption of heat by the not thoroughly heat-soaked brickwork which was used to restrict the furnace. It is to be noted that the "hydrocarbon" loss on the run of longer duration at the half pound combustion rate was nearly 1 per cent below that on the briefer run made under otherwise similar conditions.

The principal losses chargeable to the B.W.H.S. and S.H.S. were with the dry smoke pipe gases other losses chargeable to the B.W.H.S. and S.H.S. being practically negligible. Loss due

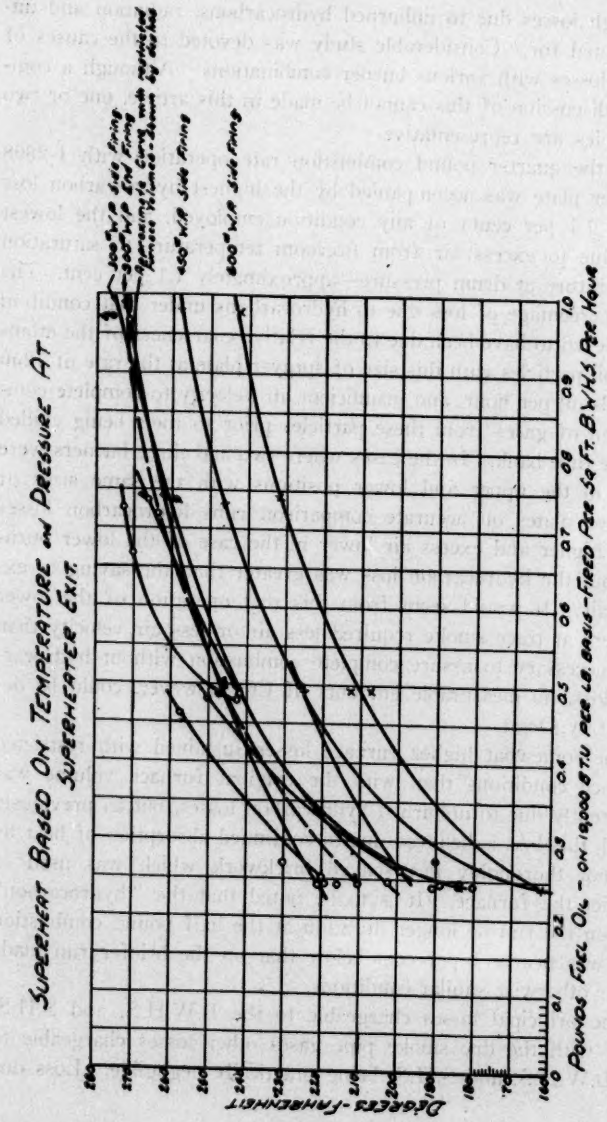

SUPERHEAT BASED ON TEMPERATURE and PRESSURE AT SUPERHEATER EXIT.
DEGREES-FAHRENHEIT
POUNDS FUEL OIL - ON 19,000 B.T.U PER LB. BASIS - FIRED PER SQ FT. B.W.H.S PER HOUR
450# W.P. Side Firing
600# W.P. Side Firing

FIGURE 28.

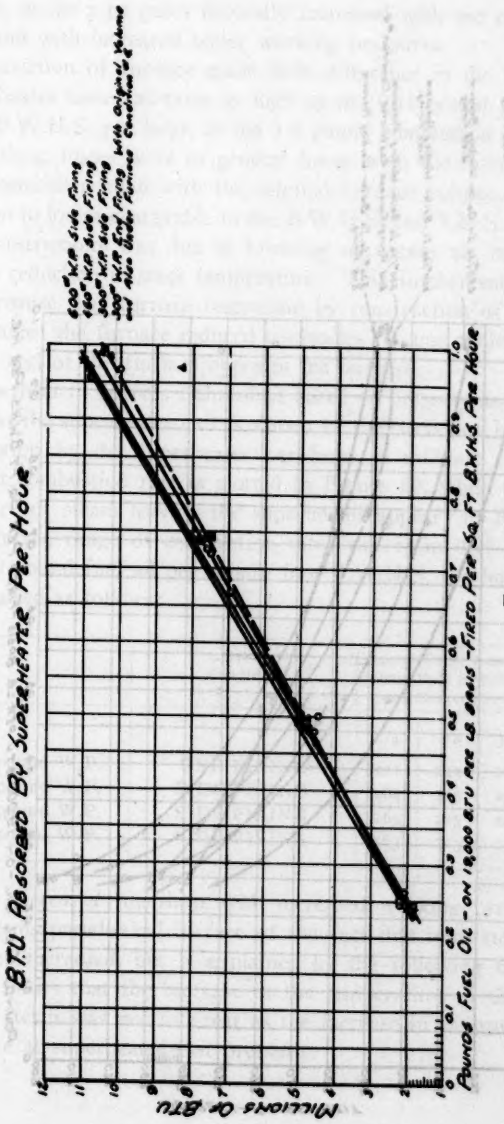

B.T.U Absorbed By Superheater Per Hour
Millions Of B.T.U.
Pounds Fuel Oil - on 18,000 B.T.U per lb basis - Fired Per Sq. Ft. B.W.S. Per Hour

FIGURE 29.

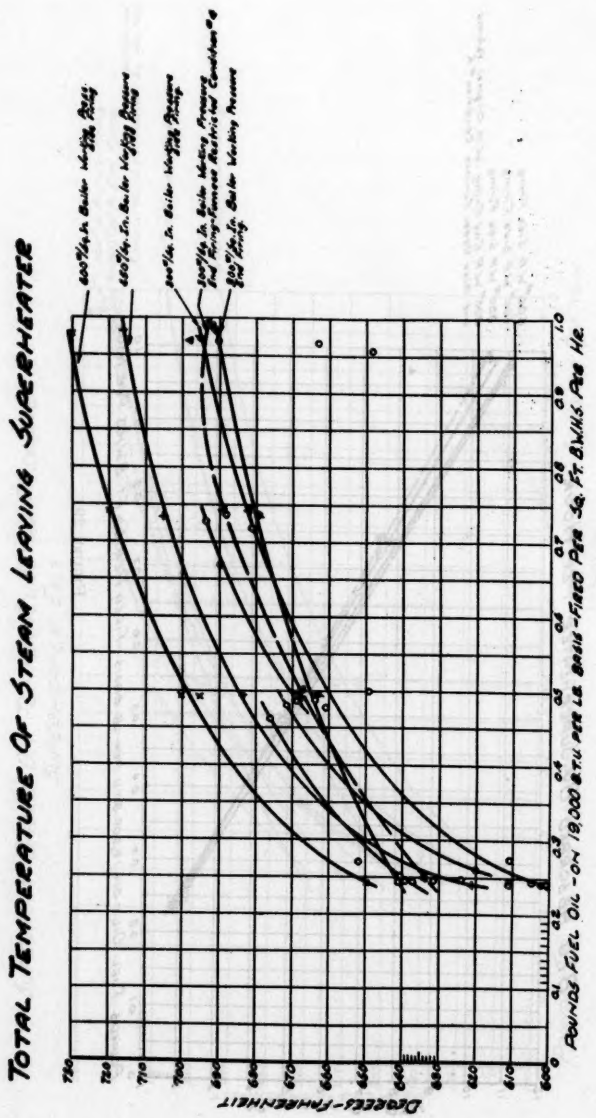

TOTAL TEMPERATURE OF STEAM LEAVING SUPERHEATER
DEGREES-FAHRENHEIT
POUNDS FUEL OIL-OR 19,000 B.T.U. PER LB. BASIS-FIRED PER SQ. FT. B.W.H.S. PER HR.

FIGURE 30.

to dry smoke pipe gases naturally increased with the combustion rate and with increased boiler working pressures.

Restriction of furnace made little difference in the boiler and superheater losses at rates as high as the 0.75 pound per square foot B.W.H.S. per hour, at the 1.0 pound combustion rate, however, these losses were in general lower with the restricted furnace conditions than with the original furnace volume. The reduction in losses chargeable to the B.W.H.S. and S.H.S. with furnace restrictions was due to lowering of excess air rather than to the reduction in stack temperature. This further substantiates the premise that furnace restriction by construction of slopes in the rear of the furnace reduced tendencies for gas eddies to form in the end of the furnace opposite the burners.

Superheat in degrees Fahrenheit based on pressure and temperature at the superheater exit is shown by the curves of Figure 28; absorption by the superheater per hour in millions of B.T.U. against combustion rate is plotted in Figure 29, while total temperature of steam leaving the superheater appears in Figure 30.

Over the range of combustion rates run from 0.25 pound to the 1.0 pound fuel oil per square foot B.W.H.S. per hour superheat varied as follows:

	Condition	Approximate Rate Pounds per Square Foot B.W.H.S. per Hour and Degrees F. Superheat			
		1/4	1/2	3/4	1
300 pound W.P.	END FIRING	221	242	264	271
300 pound W.P.	SIDE FIRING	184	243	259	276
450 pound W.P.	SIDE FIRING	163	225	247	257
600 pound W.P.	SIDE FIRING	163	213	232	243

The *decreased* superheat with increased working pressure, at first seems paradoxical in face of the fact that total steam temperature *increased,* but is explained by the following tabulation which shows that the increase in the temperature of the superheated steam was not as great as the increase in saturation temperature at superheater exit pressure.

	All Side Firing Conditions	Temperature of Steam at Superheater Exit	Increase in Temperature Over That at 300 Pounds	Saturated Temperature at Superheater Exit Pressure	Increase in Saturation Temperature at Superheater Exit Due to Pressure Above That at 300 Pounds
300 pounds per square inch gage	(1/4)	602		418	
	(1/2)	663		420	
	(3/4)	679		420	
	(1)	695		419	
450 pounds per square inch gage	(1/4)	621	19	458	40
	(1/2)	683	20	458	38
	(3/4)	705	26	458	38
	(1)	716	21	459	39
600 pounds per square inch gage	(1/4)	650	48	487	69
	(1/2)	700	37	487	67
	(3/4)	720	41	488	68
	(1)	730	35	487	68

Curves showing percentage of total heat absorption of B.W.H.S. and S.H.S. taken up by the superheater under the various conditions of operation are shown in Figure 29.

These curves show that:

Absorption by the superheater alone as compared with the boiler and superheater combined ranged as follows:

Condition	Approximate Rates per Square Foot	B.W.H.S. per Hour
	1/4 pound	1 pound
*300 pound W.P. END firing	11 per cent	13 per cent
300 pound W.P. SIDE firing	9.8 per cent	13.25 per cent
450 pound W.P. SIDE firing	10.5	13.7
600 pound W.P. SIDE firing	10.5	13.7

* During Four Hour Runs.

Use of inclined restrictions in the end of the furnace opposite the burners tended to increase the absorption by the superheater at rates above the 1/4 pound.

Under all conditions which the boiler was operated on this test, the percentage of heat absorbed by the superheater alone as compared with the boiler and superheater combined AT ANY GIVEN COMBUSTION RATE did not vary more than 1.5 per cent.

The greater percentage of heat absorbed at the 1/4 pound combustion rate 300 pounds working pressure end firing than at the same rate and pressure side firing is believed due to the closeness of the upper burners to the bank and therefore to the superheater.

OPERATION.

The boiler is a fast steamer — steam can be raised on this boiler as rapidly as on "A" type Express boilers.

The small diameter of the steam drum (42 inches) made water level more sensitive than with drums of larger diameter. Therefore, water tending was more difficult than when larger drums are used.

At no time during the operation of the boiler was priming in evidence. Even during test operation with 273 grains per gallon of chlorine present, together with 16 grains per gallons of sulphates and 14 grains of alkalinity practically no bobbing of water in the gage glasses was observed.

It is to be remembered that construction of this boiler and the course of circulation in it which is downward through the vertical headers through the tubes, upward through the diagonal headers, and thence to the steam drum, makes it to a considerable degree less susceptible to priming and geyser action than the "A" type Express boilers.

It was necessary to use superheater vents while raising steam, increasing steam pressure, and while securing the boiler in order to avoid excessive superheat and possible overheating of the superheater.

Steam entered and left the superheater at opposite ends of a single drum which was partitioned at mid-length. The difference in temperature existing between the entrance and exit ends of this drum and the tubes on the entrance and exit sides, causes considerable strain to be exerted on the superheater. This was evidenced by occasional slight leakage, developing at the first row tube ends near the drum, necessitating rerolling.

The superheater lay between the fifth (first five rows of boiler tubes are rolled into the lower drums) and the sixth row of boiler tubes (rolled into the headers). In this position, the temperature of steam leaving the superheater varies with minor changes in steam demand which are not simultaneously accompanied by similar changes in combustion conditions.

The superheater was removed at the close of this test. Inspection prior to its removal showed that some of the tubes had sagged down to the soot blower elements between the superheater and the tubes below it. The soot blowers had prevented the tubes from sagging still further. The soot blower elements were then removed and the tubes of the superheater sagged until the tubes on the superheater exit side lay between the boiler tubes below them. This sagging amounted to about 4 inches on the lowest row of superheater tubes on the exit side of the boiler, the amount of sagging tapering off to practically nil at the entrance side of the superheater.

The permanent distortion and sag of the superheater tubes was due to overheating or to difference in temperature between entrance and exit tube banks of the superheater, in spite of great precautions taken to prevent it by venting the superheater in raising and lowering the steam pressure. Several times while burner combinations and combustion rates were being altered, steam temperature leaving superheater rose as high as eight hundred, particularly during operation at 600 pounds W.P., when temperature twice rose to approximately 855 degrees F. for very brief intervals, although the steam temperatures were rapidly reduced each time this occurred. It is to be remembered that the tensile strength of boiler steel falls rapidly at high temperatures and, as the tubes themselves were undoubtedly at considerably higher temperatures than this, for brief intervals, cause of the sagging might well be traceable to this source. In other words, the high temperature caused CREEP in the steel of the superheater tubes.

Inspection of the boiler during the Shipping Board series of 425 pounds working pressure after operation at high salt concentration failed to disclose any marked sagging, but this inspection was incomplete and made from the low temperature side of superheater while all soot blowers were in place.

Inspection of the boiler after run No. 27 showed that two of the studs in one of the manhole plates were in poor condition, one stud being sheared off and the other stretched. New studs were installed and the broken ones were forwarded to the Babcock and Wilcox Company for study. In the opinion of the manufacturer, stud nuts had been taken up too tightly when the boiler was hot, and were stressed beyond their elastic limit when the boiler cooled.

In spite of the furnace insulation boiler casings became very hot, and it was thought advisable to install additional exterior insulation outside the boiler casing.

Inspection of the furnace both after run No. 58 and again after No. 84 showed the furnace brickwork in good condition. Certain contraction cracking had occurred, however, and moderate slag deposition, to the depth of about one-half inch, had occurred on the refractory surface throughout the furnace.

After run No. 58 it was found necessary to install a flexible copper bulkhead joint in order to allow movement of the steam line. The strain during sudden contractions of the steam line was otherwise transmitted to the studs securing steam line to the superheater exit flange. Use of H.G. specification studs and a Flexitallic gasket at this joint undoubtedly minimized the difficulty experienced with it.

Flexible shaft tube cleaners were required for cleaning tubes, and flexible shaft rollers for rerolling them.

The total number of steaming hours during which this boiler was in operation exclusive of runs without superheater and safety valve tests conducted after the series were 535.

The boiler can be operated at high overall (boiler, furnace and burners) efficiency over the range of rates from the quarter to the one pound per square foot per hour when fired either from the HIGHER END or SIDE of the furnace.

Fireroom air pressures required under END FIRING conditions were excessive at rates above the 0.5 pound per square foot B.W.H.S. per hour, pressures in the vicinity of 12 inches (of water) being necessary at the 1.0 pound combustion rate.

Fireroom air pressures required under SIDE FIRING conditions were less than with END FIRING because a larger number

of burners and hence a lower rate per burner could be employed; pressures in the vicinity of 8.5 inches (of water) were necessary, however, for the 1.0 pound combustion rate.

Overall efficiencies obtained with constricted furnace volumes were inferior to those with the original furnace volume. Fireroom efficiencies were also inferior except at the 1.0 pound combustion rate where they were slightly higher. Best results with constricted furnace were obtained with conditions 3 and 4 when inclined surfaces were used in the rear of the furnace. Results indicate that, had runs been of sufficient duration to thoroughly heat soak the brickwork used for restriction, higher efficiencies would have been obtained, and also, therefore, that results would have been superior if the boiler were constructed with constricted furnace volume, but normal thickness of brickwork.

A heat release of over 317,000 B.T.U. per cubic foot furnace volume per hour was obtained without difficulty — overall boiler efficiency being 82.1 per cent — furnace and burners efficiency as calculated exceeding 97 per cent.

Operation of the boiler at 450 and 600 pounds per square inch gage working pressure was not more difficult than at 300 pounds per square inch gage working pressure except for the necessity of exercising greater care in venting superheater when increasing and decreasing pressure and greater difficulty in controlling steam pressure by bleeder.

CONCLUSIONS.

The boiler is of sturdy construction. It is a rapid steamer.

Circulation in the boiler is good and is not conducive to priming even with high salt concentrations. The boiler can be fired satisfactorily either from the higher end of the furnace or the side at combustion rates up to 1.0 pound per square foot B.W.H.S. per hour.

It can be safely and satisfactorily operated at pressures up to 600 pounds per square inch gage and at combustion rates up to 1.0 pound per square foot B.W.H.S. per hour.

Overall efficiencies obtainable are high, but their value is somewhat reduced by high fireroom air pressure requirements, due to gas path restriction and number of tube rows.

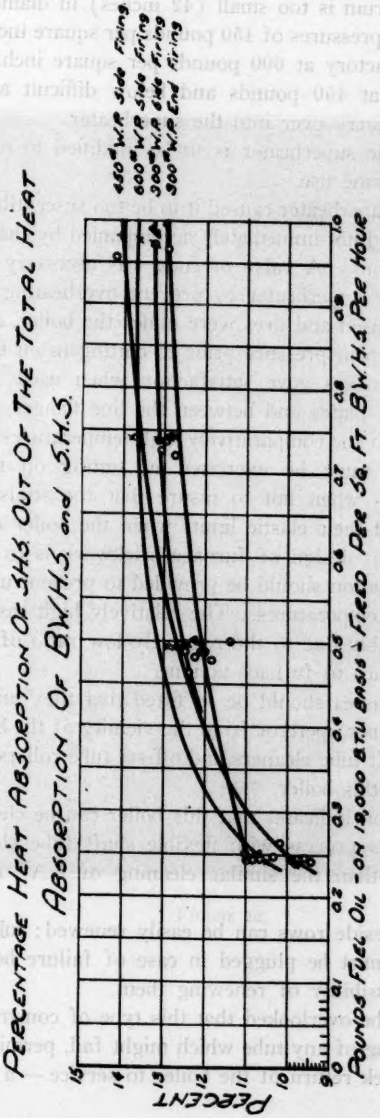

PERCENTAGE HEAT ABSORPTION OF SH'S OUT OF THE TOTAL HEAT
ABSORPTION OF B.W.H.S. and S.H.S.
450# W.P. Side Firing
300# W.P. Side Firing
300# W.P. Side Firing
300# W.P. End Firing
PERCENT
POUNDS FUEL OIL - ON 18,000 B.T.U. BASIS - FIRED PER SQ. FT. B.W.H.S. PER HOUR

FIGURE 31.

The steam drum is too small (42 inches) in diameter for ease of operation at pressures of 450 pounds per square inch and below, although satisfactory at 600 pounds per square inch. It renders water tending at 450 pounds and below difficult and increases probability of carry over into the superheater.

Design of the superheater is to be modified to make it satisfactory for marine use.

Position of superheater caused it to be too susceptible to changes in steam demand not immediately accompanied by changes in combustion conditions. A valve or cock was necessary for bleeding steam from the superheater to prevent overheating when steam was not being used and fires were under the boiler, as is the case when coming up to pressure prior to cutting in on the line.

Flexitallic gaskets gave satisfaction when used for handhole plates, manhole plates and between the line flanges.

On account of the comparatively high temperatures at the higher pressures care must be exercised in setting on manhole and handhole plates when hot to insure that the studs will not be stressed beyond their elastic limits when the boiler cools.

If the present design of furnace brickwork is to be employed additional insulation should be provided to prevent uncomfortably high fireroom temperatures. The relatively high casing temperature was probably due to the relatively low ratio of radiant heat absorbing surface to furnace volume.

The gage glasses should be so fitted that they may be readily seen by the burner operator from the vicinity of the burners.

Flexible shaft tube cleaners and off-set tube rollers are essential equipment for this boiler.

Demonstration indicates that this boiler can be cleaned without difficulty. This process with flexible shaft tube cleaners is not more difficult than the similar cleaning of "A" type Express boilers.

Tubes in fireside rows can be easily renewed; tubes above the fireside rows must be plugged in case of failure because of the practical impossibility of renewing them.

It is not to be overlooked that this type of construction allows prompt plugging of any tube which might fail, permitting a correspondingly quick return of the boiler to service — a distinct mili-

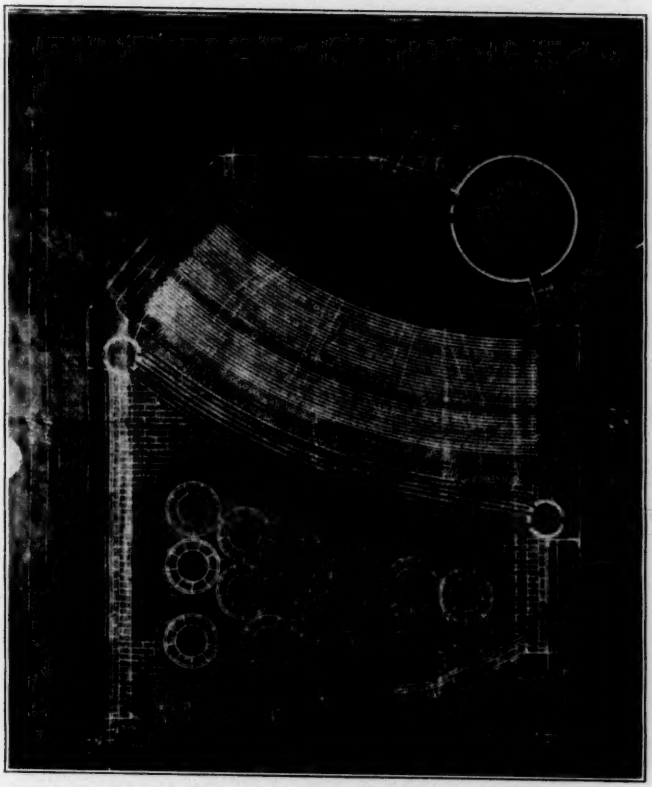


FIGURE 32.

tary advantage. Plugging of tubes in "A" type Express boilers necessitates entering the drums, consequently boilers of that type must be allowed to cool before tube plugging can be begun.

Twenty-four boilers of the Sectional Express type, but for 300 pounds working pressure are now under construction for the light cruisers *New Orleans* (32), *Astoria* (34) and *Minneapolis* (36); other boilers of the same type also will be built for additional cruisers of this class. The parts for these boilers are being manufactured by the Babcock and Wilcox Company, but they will be assembled for each ship by the Navy Yards.

These boilers embody design changes which this test showed desirable, particularly in steam drum diameter, superheater arrangement and steam baffling. They are of ten thousand nine hundred square feet boiler water heating surface and nine hundred eighteen square feet superheater heating surface. Although the boilers will be fired from the side with twelve Cuyama burners floors will have curved slopes somewhat similar to that used during furnace restriction condition No. 3.

One of these boilers is now being erected at the Fuel Oil Testing Plant. It will be subjected to test operation in the near future, and then be installed in Cruiser No. 36, U.S.S. *Minneapolis*. Side elevation of this boiler is shown in Figure 32.

FIREBRICK FOR NAVAL BOILERS.

By James B. Donnelly, Lieutenant, U. S. N.

In the last few years a great amount of work has been done by various investigators in the study of properties of refractories and the service conditions that influence their performance. As far as boiler furnace refractories are concerned, most of this work has been done in connection with performance of refractories in stationary boiler furnaces. The operating conditions in Naval boiler furnaces that affect refractory linings are somewhat different from those in stationary boiler furnaces. As this difference is principally one of degree much of the knowledge of factors that cause failure of refractories in stationary practice is applicable in a general way to our own case.

While several forms of refractory materials are used in Naval boiler furnaces, the major part of the furnace lining consists of fire clay brick and the life of the lining is determined primarily by the performance of these brick. Some knowledge of the make up of a fire clay brick is essential to an understanding of its behavior in the boiler furnace. As its name indicates the fire clay brick is made of fire clay. Fire clay is the most common of the refractory materials, being found generally over the whole world. It is a product of nature formed by the decomposition of rock by weather and heat. As might be expected it is a rather complicated material. The most important constituent of clay is the so-called "clay base." This is an hydrated silicate of alumina, either kaolinite $Al_2O_3.2SiO_2.2H_2O$ or hydromica which has the same proportions of silica and alumina with some of the water replaced by alkali. Mixed with the clay base are a number of other materials such as free silica and alumina, and the so-called impurities such as rutile (TiO_2), the alkalies, soda and potash, compounds of iron, magnesium and lime and organic matter. Typical analyses of fire clays are shown in Table 1. It is cus-

tomary to determine the content of the various elements as if they were present in the form of oxides, although in the material they do not necessarily occur as such. A rational analysis is too difficult and expensive to make and is not particularly accurate. It is noted that the combined silica and alumina content makes up about 80 to 85 per cent of the whole clay.

TABLE I—TYPICAL ANALYSES OF FIRE CLAY.

	Per cent	Per cent	Per cent
Loss on Ignition	11.1	10.7	9.60
Silicon Dioxide	51.0	50.2	60.40
Aluminum Oxide	32.5	34.3	31.00
Iron Oxide	1.7	1.8	3.28
Titanium Oxide	1.7	2.2	2.84
Calcium and Magnesium Oxide	0.2	0.3	.67
Alkali as Potassium Oxide	1.8	0.5	1.95
P. C. E. Cone No	31	31–32	26–27
	°F.	°F.	°F.
	3065	3065–3101	2912–2948

The physical properties that are peculiar to fire clay are: high softening temperature, plasticity, property of burning to a clinker, etc. The high softening temperature is the property that distinguishes the fire clays from the cheaper clays such as those used in making common brick. Fire clay does not have a definite melting point but gradually softens and melts over a range of temperature. In order to be able to compare softening temperatures of different clays this property is determined in terms of P.C.E. (Pyrometric Cone Equivalent). Orton cones, used in this determination, are small pyramids about two inches high which are made of special materials and which soften at certain fixed temperatures designated by the number of the cone, *e.g.*, Cone No. 30 softens at 3038 degrees F., No. 32 at 3101 degrees F., etc. Cones of the test material and various numbered Orton cones are heated together in a furnace until the test cones soften. The highest numbered Orton cone which has at this time softened determines the P.C.E. of the test material.

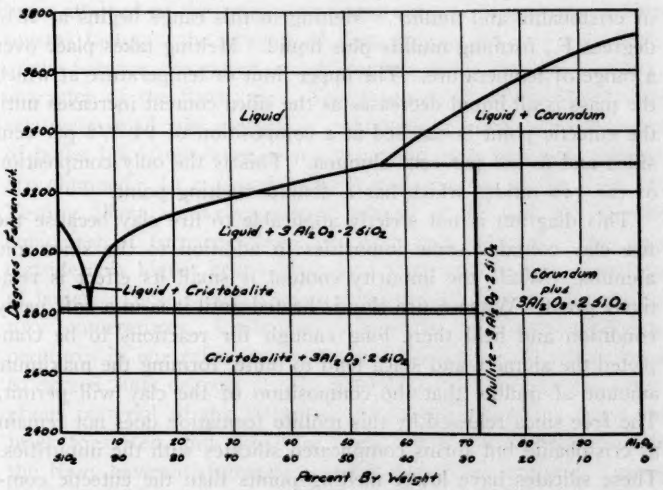
Based on their physical properties fire clays may be classified as flint clays or plastic clays. The flint clays are hard, have comparatively high softening temperatures and have little or no plas-

ticity. Their plasticity cannot be developed by tempering. The plastic clays are softer than the flints, generally have lower melting points, have a greater degree of plasticity and their plasticity can be developed by tempering. There is no abrupt line of demarcation between the two. Those clays having intermediate properties are called semi-flints or semi-plastics as the case may be. In making firebrick plastic clay is added to flint clay, or to a mixture of flint and semi-flint, in order to give the mass the necessary plasticity for working and to provide a bond so that brick can withstand the necessary handling prior to burning. The firebrick is then a mixture of two or more clays. There are a number of processes employed in the manufacture of the brick. They vary in detail with the properties that are desired in the finished brick. In general the various ingredients are ground to the necessary fineness, thoroughly mixed, tempered with water, moulded and fired in a kiln. In the process of burning the water of formation and the combined water are driven off, the organic matter is burned off and certain mineralogical changes take place. During the final stages of burning some of the material of the brick melts and begins to fill the pores. This is known as vitrification. In the early stages of vitrification the grains of the brick have softened just enough to be cemented together. When sufficient of the material has fused to completely fill the pores complete vitrification is said to have taken place. The expelling of the water, burning off of the organic matter and the vitrification of the clay allows the particles of the brick to pack closer together resulting in shrinkage. Calcined clay is sometimes used as one of the ingredients of the mixture in order to reduce this shrinkage. The finished brick is a strong, hard, moderately porous mass consisting of grains of the various clays bonded together by the glass or fused material produced by vitrification. Typical analyses of fire clay brick are given in Table II. It is noted that they do not differ much from those of raw fire clay except that due to loss of water and organic matter the combined silica and alumina now make up 90 – 95 per cent of the total. Some changes in structure have undoubtedly taken place during the burning of the brick but these are not indicated by the chemical analysis.

TABLE II—TYPICAL ANALYSES OF FIRE CLAY BRICK.

	Per cent	Per cent	Per cent
Loss on Ignition	0.1	0.1	0.1
Silicon Dioxide	54.4	52.8	61.0
Aluminum Oxide	40.3	42.4	35.0
Titanium Oxide	2.5	1.6	1.1
Iron Oxide	1.5	2.1	2.2
Calcium and Magnesium Oxide	0.1	0.2	0.1
Alkali as Potassium Oxide	1.1	0.8	0.5
P. C. E. Cone No.	33	31	32
	°F.	°F.	°F.
	3128	3065	3101

Since silica and alumina make up by far the greater part of the brick the relations between these two oxides at high temperatures are important. The equilibrium diagram for the silica-alumina system as determined by Bowen & Greig is shown in Figure 1. At the temperatures involved in this diagram there is but one stable compound of silica and alumina. This com-



EQUILIBRIUM DIAGRAM OF THE SYSTEM $Al_2O_3 - SiO_2$ (BOWEN and GREIG)

FIG. 1.

Reproduced with permission from E. B. Powell's paper "Boiler Furnace Refractories" published by A. S. M. E.

pound is called mullite, has the chemical formula $3Al_2O_3.2SiO_2$ and contains 72 per cent by weight of alumina and 28 per cent of silica. Free silica appears as cristobalite while free alumina appears as corundum. For all compositions of the mixture in which the alumina content is greater than it is in mullite melting begins at 3290 degrees F., forming liquid plus corundum. Melting takes place over a range of temperature and the temperature at which the mass is all liquid increases as the alumina content is increased. At temperatures below 3290 degrees F. the structure is mullite plus corundum. At the composition of mullite melting also begins at 3290 degrees F., forming liquid plus corundum in the melting range. Below 3290 degrees F., however, the mass consists solely of mullite crystals. For the average fire clay the alumina content is considerably below that of mullite (72 per cent). In this range of compositions the silica content is in excess of the amount required for combination with the alumina to form mullite. At temperature below that at which melting begins the excess silica appears as cristobalite and the structure is made up of cristobalite and mullite. Melting in this range begins at 2813 degrees F., forming mullite plus liquid. Melting takes place over a range of temperature. The upper limit or temperature at which the mass is all liquid decreases as the silica content increases until the eutectic point is reached at a composition of 94 1/2 per cent silica and 5 1/2 per cent alumina. This is the only composition of the two oxides which has a definite melting point.

This diagram is not strictly applicable to fire clay because the fire clay contains some impurities in addition to the silica and alumina. While the impurity content is small its effect is relatively great. When a fire clay is heated until it is in a soft pasty condition and held there long enough for reactions to be completed the alumina and silica tend to unite, forming the maximum amount of mullite that the composition of the clay will permit. The free silica released by this mullite formation does not remain as cristobalite but forms complicated silicates with the impurities. These silicates have lower melting points than the eutectic composition of silica and alumina and their melting points decrease as the number and quantity of impurities is increased. For a clay of given silica-alumina ratio the melting temperatures will then

be lower than those shown in Figure 1 by an amount depending on the number and amount of impurities present. The structure of a clay which has been heated to high temperatures for long periods will be mullite crystals as in Figure 1 but the cristobalite will be replaced by a matrix of glass — amorphous material — made up of the silicates of the impurities.

The average firebrick is not burned sufficiently in the kiln for these reactions to be completed. Its structure will differ considerably from that of a clay burned for a long period at a high temperature. Some mullite will be formed in the brick but in no particular arrangement or dispersion as its formation is incidental to and not an object of the burning. The brick will contain several materials in addition to mullite and the amorphous material will be considerably less in quantity and different in composition.

TEMPERATURE.

One of the most important properties that a brick must have is ability to withstand the effects of high temperature. The temperatures attained in the refractory lining of any furnace depend on several factors: the amount of heat release, the size and shape of the furnace, the extent of cold surface in the furnace envelope, character of the flame, etc. The furnaces of Naval boilers are relatively small; the amount of cold surface varies with the type of boiler but is not large in any type; and very high heat releases are obtained. In a Naval boiler tested recently at the Fuel Oil Testing Plant a maximum heat release of 317,500 B.T.U. per cubic feet of furnace volume per hour was obtained. This is a considerably higher heat release than is secured in any other type of boiler. These factors tend to produce high furnace and refractory temperatures. Little data is available as to the actual temperatures in our refractory walls in service but it is not believed to exceed 3000 degrees F. At such high temperatures the amorphous material of the brick may be liquid enough to cause the brick to soften, and distort or run. All fire clay brick used in the Navy have satisfactorily passed a test at the Fuel Oil Testing Plant at a temperature of 3000 degrees F. Failure of brick in service by fusion or softening should therefore be rare. It is believed that in most cases where fusion of the brick has been reported the real cause of failure was slag action.

Temperature has several effects on the brick other than that of softening or fusion. One of these is thermal expansion. When a brick is heated a more or less regular reversible thermal expansion takes place. Some of the materials in the brick, notably free silica, exist in two or more allotropic forms which are stable over different temperature ranges. Certain of the inversions or changes in allotropic form are accompanied by a marked volume change so that in heating the brick through these ranges irregularities in expansion are introduced, the extent of which depends on the amount of the material in question that is present. The expansion characteristics of some different types of brick as determined by Professor F. H. Norton[1] are shown in Figures 2 and 3. In Figure 2 the expansion curve of a silica brick containing about 95 per cent silica shows the irregularities introduced by silica inversions. In the expansion curves for kaolin brick shown in the same figure some slight irregularities can be seen at the same temperatures as those in the silica brick. These are apparently due to inversions of small amounts of free silica. Kaolin is the name given to a particular fire clay that has a very low impurity content. The curves for kaolin brick show the effect of burning the brick at different temperatures. In all cases a more or less regular reversible expansion takes place as the temperature is increased until a critical temperature is reached. As the tempera-

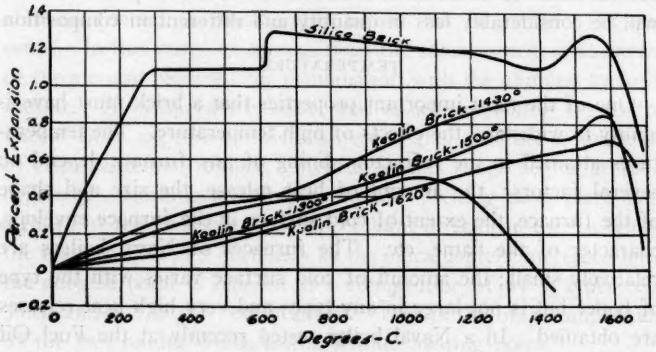


FIG. 2.
Drawn with permission from Curves by F. H. Norton.

ature of burn is increased the critical temperature is raised as well as the softening temperature, and they approach each other. The curves for Missouri fire clay brick and Colorado fire clay brick in Figure 3 show in general the same characteristics as do those for kaolin brick. In the curves for Pennsylvania and Maryland fire clay brick a permanent expansion is exhibited beginning at the critical temperature and continuing up to the softening point. This expansion, called secondary expansion, is peculiar to some clays and is thought to be due to reaction of the fluxes liberating gases which bloat the structure. There seems then to be a critical temperature for any fire clay brick above which some permanent volume change will occur. The secondary expansion, that is peculiar to some bricks, does not take place if the brick is under pres-

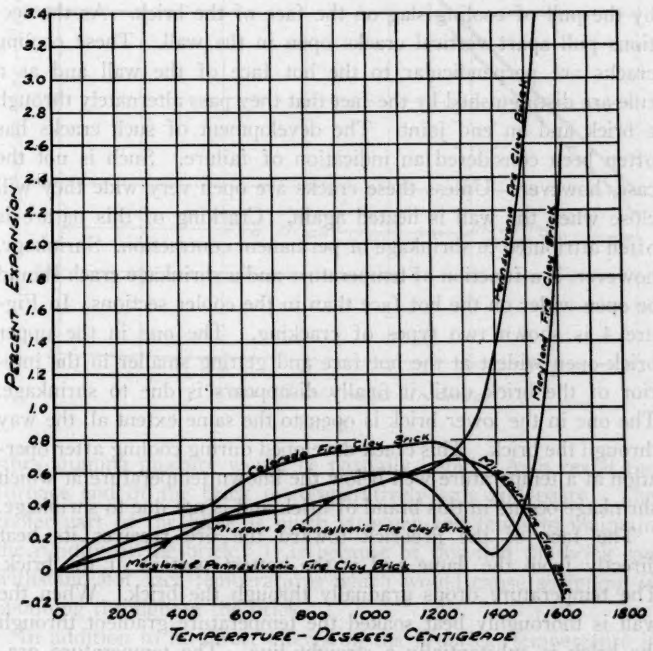


FIG. 3.

Drawn with permission from Curves by F. H. Norton.

sure such as it is in a furnace wall. The liberation of gases is manifested, however, by the formation of bubble-like pores in the brick.

When the furnace wall is heated each brick in expanding pushes against the adjacent bricks and the wall expands as a unit. On cooling the action is somewhat different. If there were no restraining force each brick would contract in place and leave all the vertical joints open. The weight of the wall is, of course, sufficient to cause the wall to contract as a unit in the vertical direction so that the horizontal joints always remain closed. However, the temperature over the furnace wall during operation varies considerably and the rate at which the various parts of the wall cool down also varies. Unequal cooling of sections of the wall tend to make them pull apart during contraction. This action is increased by the pull of cooling slag on the face of the brick. As the sections pull apart vertical cracks open in the wall. These cooling cracks are perpendicular to the hot face of the wall and as a rule are distinguished by the fact that they pass alternately through a brick and an end joint. The development of such cracks has often been considered an indication of failure. Such is not the case, however. Unless these cracks are open very wide they will close when the wall is heated again. Cracking of this nature is often attributed to shrinkage or permanent contraction. Shrinkage, however, is a function of temperature and a shrinkage crack should be open wider on the hot face than in the cooler sections. In Figure 4 is shown two types of cracking. The one in the upper brick open widest at the hot face and getting smaller in the interior of the brick until it finally disappears is due to shrinkage. The one in the lower brick is open to the same extent all the way through the brick. This crack developed during cooling after operation at a temperature well below the known temperature at which shrinkage occurs in this brand of brick and is not due to shrinkage.

The face of the firebrick toward the fire receives its heat directly from the flame and is the hottest portion of the brick. The temperature drops gradually through the brick. When the wall is thoroughly heat soaked the temperature gradient through the brick is substantially a straight line. The temperature gradients, shown in Figure 5, were obtained on walls of a test fur-

nace which were backed up by the same insulation as is used in our furnaces. The gradient line is shown for the firebrick and the first course of insulation only. The slope of the gradient line through the firebrick of course depends on the amount of insulation behind the firebrick. With perfect insulation it would be a horizontal line. It is to be noted that the temperatures in the hot end are in the range where the structure of even a pure

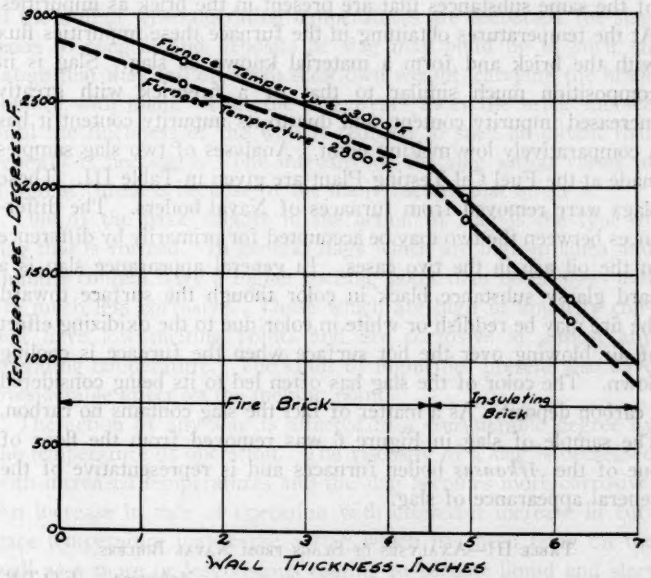


FIG. 5.

silica-alumina mixture would be partially liquid. As a result the furnace end of the brick is comparatively soft and pasty. The cooler part of the brick is much stronger and tends to maintain the rigidity of the brick. It is because of this that the brick can withstand hot face temperatures which would cause softening if obtaining throughout the brick.

In addition to its direct effects on the firebrick, temperature is an important factor in the two major causes of brick failures in our boiler furnaces: slagging and spalling.

SLAGGING.

Practically all fuels contain infusible ingredients which are designated by the general term ash. In our oil-fired boilers all of the ash is passed into the furnace with the fuel where it is heated to a high temperature. In the course of its passage through the furnace some of this hot ash impinges on the soft pasty face of the brick where it tends to adhere. The ash contains many of the same substances that are present in the brick as impurities. At the temperatures obtaining in the furnace these impurities flux with the brick and form a material known as slag. Slag is in composition much similar to that of a firebrick with greatly increased impurity content, and due to its impurity content it has a comparatively low melting point. Analyses of two slag samples made at the Fuel Oil Testing Plant are given in Table III. These slags were removed from furnaces of Naval boilers. The differences between the two may be accounted for primarily by difference in the oil ash in the two cases. In general appearance slag is a hard glassy substance black in color though the surface toward the fire may be reddish or white in color due to the oxidizing effect of air blowing over the hot surface when the furnace is cooling down. The color of the slag has often led to its being considered a carbon deposit. As a matter of fact the slag contains no carbon. The sample of slag in Figure 6 was removed from the floor of one of the *Arkansas* boiler furnaces and is representative of the general appearance of slag.

TABLE III—ANALYSES OF SLAGS FROM NAVAL BOILERS.

	Arkansas Boiler Per cent	F.O.T.P. Boiler Per cent
Loss on Ignition	0.3	
SiO_2	44.3	44.4
Al_2O_3	25.7	36.3
Fe_2O_3	8.6	11.9
TiO_2	3.1	2.5
CaO	2.2	6.5
MgO	1.8	0.4
Na_2O plus K_2O	12.4	0.5

Particles of the brick surface are absorbed in the process of slag formation. As the formation continues the slag penetrates into the brick and further absorption of brick particles takes place. Due to its low melting point the slag is in a more or less liquid state at normal operating temperatures. If the slag is liquid enough, it will run over the face of the wall with a dissolving action, leaving a rough eroded brick surface behind it for fresh slag formation. When the running slag reaches cooler sections of the wall or when operating temperatures are decreased, the slag tends to congeal and deposits of slag may build up to such an extent that they fall off from their own weight, carrying the brick surface with them. Thus the slag wears away the brick surface until the thickness of the wall is reduced to such an extent that renewal is necessary. The intensity of the slag action varies in different furnaces because of the many factors that affect it.

One of the basic factors in the action of slag is the type of slag that is formed. In general slags which are high in silica and alumina content have a higher melting point than other slags and are much less corrosive. Those which are high in impurity content have low melting points and are corrosive at almost any operating temperature. The kinds of impurities present also have considerable effect on the melting point.

The action of any slag is affected to a considerable degree by the temperature of operation. The viscosity of a slag is decreased with increased temperatures and the slag becomes more corrosive. An increase in rate of operation with attendant increase in furnace temperature may cause a slag which has been lying on the wall as a more or less viscous coating to become liquid and start running over the wall, while a decrease in steaming rate may cause a liquid slag to become so viscous that it does not run at all. The extent of the effect of temperature on some coal slags as determined by Professor R. K. Hursh [2] is shown in Figure 7. Oil slags should follow the same general trends though perhaps not to as great a degree. It is noted, in a number of cases, that up to a certain temperature the erosion loss increases regularly as the temperature increases. Above this temperature the rate of erosion loss increases very much more rapidly for a given temperature increase. Klinefelter and Rexford [3] have recently stated that

they found a critical temperature for every slag examined. Above this critical temperature the slag was particularly corrosive. Below it the slag was much less corrosive and corrosiveness decreased with decreasing temperature.

While every effort is made to secure a complete combustion, in no furnaces does it take place immediately. Consequently there are at least parts of the furnace where the atmosphere is probably reducing, *i.e.*, seeking oxygen. A reducing atmosphere in vicinity of slags will reduce the iron compounds in the slag from the ferric state to the ferrous or ferroso-ferric state. The ferrous slags have a much lower melting point and are much more corrosive than the ferric slags. At the operating temperatures of modern boiler furnaces ferric compounds break down as a temperature effect. In such cases the furnace atmosphere has little or no effect on the slag action. At lower operating temperatures, however, the furnace atmosphere may have considerable effect on the action of the slag.

The rate at which slag forms depends on the rate of renewal of the slag forming constituents. Fuel oil used in the Navy has a very low ash content, ranging from .02 to .07 per cent by weight. On account of this low ash content many people have been led to believe that slag did not form in our furnaces. We do, however, fire large quantities of fuel and an appreciable amount of ash can be passed into the furnace in a comparatively short time. Assume a boiler with 12,000 square feet of heating surface steaming at the pound rate for ten days with oil of .04 per cent ash content. During this period about 1152 pounds of ash will have been passed into the furnace. A comparatively small portion of this deposited on the brick surfaces of our small furnace would form an appreciable amount of slag. The rate of formation is, of course, much slower in our boiler furnaces than is the case with fuels of higher ash content such as coal, but it does nevertheless form and is a factor in the failure of the refractories.

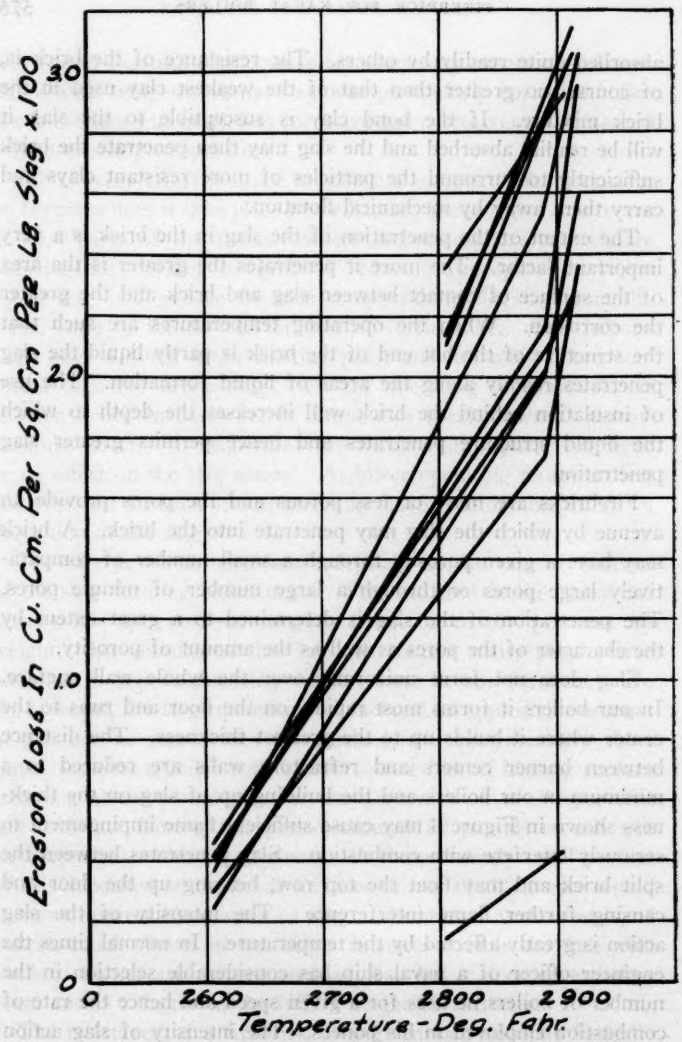
The resistance of a firebrick to a given slag depends basically on the inherent resistance of the clay or clays used in the brick but is also considerably affected by the physical properties of the brick. Both slags and clays vary widely in composition and properties. A clay may be quite resistant to some slags and

absorbed quite readily by others. The resistance of the brick is, of course, no greater than that of the weakest clay used in the brick mixture. If the bond clay is susceptible to the slag it will be readily absorbed and the slag may then penetrate the brick sufficiently to surround the particles of more resistant clays and carry them away by mechanical flotation.

The extent of the penetration of the slag in the brick is a very important factor. The more it penetrates the greater is the area of the surface of contact between slag and brick and the greater the corrosion. When the operating temperatures are such that the structure of the hot end of the brick is partly liquid the slag penetrates readily along the areas of liquid formation. The use of insulation behind the brick wall increases the depth to which the liquid structure penetrates and hence permits greater slag penetration.

Firebricks are more or less porous and the pores provide an avenue by which the slag may penetrate into the brick. A brick may have a given porosity through a small number of comparatively large pores or through a large number of minute pores. The penetration of the slag is determined to a great extent by the character of the pores as well as the amount of porosity.

Slag does not form uniformly over the whole wall surface. In our boilers it forms most rapidly on the floor and runs to the center where it builds up to the greatest thickness. The distance between burner centers and refractory walls are reduced to a minimum in our boilers and the building up of slag on the thickness shown in Figure 6 may cause sufficient flame impingement to seriously interfere with combustion. Slag penetrates between the split brick and may float the top row, heaving up the floor and causing further flame interference. The intensity of the slag action is greatly affected by the temperature. In normal times the engineer officer of a naval ship has considerable selection in the number of boilers he uses for a given speed and hence the rate of combustion employed in his boilers. The intensity of slag action will then vary considerably on different ships being a maximum on those ships which use few boilers at high steaming rates for a given speed. The author recently had occasion to examine the furnaces of a ship just returned from acceptance trials involving



EFFECT OF TEMPERATURE ON EROSION BY SLAGS.

FIG. 7.

Reproduced with permission from R. K. Hursh's paper "A Laboratory Slagging Test for Boiler Furnace Refractories," published by A. S. M. E.

several days steady operation at high rates. Although the furnace linings had been in use only about one hundred and fiftv hours thin streams of congealed slag were visible on the furnace face of the lining. In a boiler furnace of one of the Fuel Oil Testing Plant boilers, after some 850 hours of operation at varying rates a thin coating of slag was found on the face of the brick. There was, however, little or no evidence of running of the slag on the side and back walls. The periods of operation of this boiler were all comparatively short — about 6 to 8 hours — so that the walls were never heat soaked, and for a large portion of the time in use it was steamed at comparatively low rates.

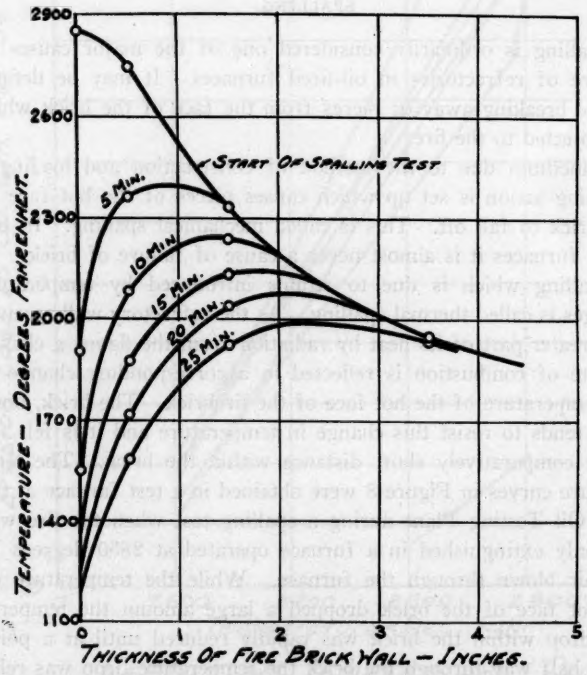
SPALLING.

Spalling is ordinarily considered one of the major causes of failure of refractories in oil-fired furnaces. It may be defined as the breaking away of pieces from the face of the brick which is subjected to the fire.

Sometimes due to the method of construction and loading a pinching action is set up which causes pieces of the hot face of the brick to fall off. This is called mechanical spalling. In our small furnaces it is almost never a cause of failure of brick.

Spalling which is due to strains introduced by temperature changes is called thermal spalling. As the refractory wall receives the greater part of its heat by radiation from the flame, a change in rate of combustion is reflected in a corresponding change in the temperature of the hot face of the firebrick. The brick, however, tends to resist this change in temperature and it is felt for but a comparatively short distance within the brick. The temperature curves in Figure 8 were obtained in a test furnace at the Fuel Oil Testing Plant during a spalling test when the fire was suddenly extinguished in a furnace operated at 2850 degrees F. and air blown through the furnace. While the temperature of the hot face of the brick dropped a large amount the temperature drop within the brick was rapidly reduced until at a point about half way through the brick the temperature drop was relatively small. When fires are lighted in the furnace again the reverse action takes place, *i.e.*, the temperature of the hot face

increases rapidly up to 2850 degrees F. again while the temperatures of the cool end remain the same. As a result the hot end goes through rapid cycles of expansion and contraction while the cool end of the brick experiences no volume change. This volume change introduces strains in the brick and if the brick is not sufficiently flexible to absorb the strains by readjustment of the particles fracture occurs and the hot face breaks off. If the temperature change involved happens to be within the range where the brick shows an irregularity in expansion the strain is further increased and liability of spalling fracture increased. Investigations by Professor F. H. Norton [4] and M. C. Booze and S. M. Phelps [5]



TEMPERATURES WITHIN BRICK AT VARIOUS INTERVALS DURING A 25 MINUTE SPALLING TEST.

FIG. 8.

indicate that the spalling resistance of a brick depends on its coefficient of expansion, heat conductivity, specific heat and flexibility. Of these, flexibility is the most important because it is capable of regulation through the widest range, while the range through which the other facts may be changed is limited.

It has long been know that brick, which before use had an excellent resistance to thermal spalling, sometimes spalled comparatively readily after they had been in service for a time. This has been attributed to structural changes that take place in the brick under service conditions and spalling due to this change is called structural spalling. It is now believed that most first grade firebrick have sufficient flexibility to withstand the strains put on it due to temperature changes that occur in service and that most of the spalling that occurs is due to or influenced to a great degree by change in structure.

While the maximum temperature attained by the face of the firebrick may not reach 3000 degrees F. it is considerably above the temperature to which the brick was burned in the kiln. Temperatures above the burning temperature also obtain for some distance within the brick. These temperatures maintained over a long period of time permit reactions that were only started in the kiln to be carried to completion. The reactions will tend to produce further crystallization of mullite. However, the composition of the average fire clay is such that mullite even if formed to the maximum amount will not make up a very large part of the structure. The principal change brought about is the production of increased amounts of amorphous material or glass and a more thorough distribution of the impurities throughout the amorphous material which may make it less viscous. The bricks shown in Figure 9 have been heated to 3000 degrees F. on the hot face for a period of only 24 hours. In the cooler part of the brick the separate grains may be distinctly seen. This part of the brick is unchanged. Nearer the hot end due to increased glass production the outline of the grains become fainter. Close to the hot face the outline of the grain has practically disappeared and to the naked eye this part of the brick is solely an amorphous substance or glass. These changes were brought about purely as a result of temperature. In Service the change in structure of the hot end is

further complicated by the penetration of slag which mixes most readily with the glass and further changes the composition of this material. The brick is then no longer homogeneous. The properties which previously gave the brick a high spalling resistance may now be so radically changed that spalling under temperature change may take place readily. The materials of which the hot face of the brick are now composed may be susceptible to thermal shock.

A temperature of 3000 degrees F. is considerably above the critical temperature for practically all firebrick so that in addition to structural change in the hot end of the firebrick we may expect some manifestation of volume change. Permanent contraction usually has one of two effects. The brick may become wedge shaped or shrinkage cracks may develop in the brick surface. The brick shown in Figure 10 has become wedge shaped from shrinkage. The bricks in Figure 11 have become wedge shaped causing opening of the joints and shrinkage cracks have also developed. Needless to say this brick was heated appreciably above its critical temperature. In the case of those brick made of clays exhibiting secondary expansion the indications of having passed the critical temperature are somewhat different. The panel of brick shown in Figure 12 was laid up with such brick. It is to be noted that the bricks have bloated somewhat, particularly near the edges, and in the surface of the brick may be seen a number of blisters and small holes probably caused by gases bubbling through the soft pasty face of the brick. In Figure 9 it may be seen that the formation of these bubble like holes penetrates for an appreciable depth in the brick.

When the volume change is marked, sufficient strain may be introduced in the brick to cause the hot end to crack off. In Figure 10 a spalling crack may be seen about an inch and a half back from the hot face. This brick had not been subjected to any temperature change other than the normal heating up and cooling down incident to the firing and securing of the furnace. The brick in Figure 13 spalled during operation of the furnace at 3000 degrees F. either due to structural change, volume change or both. Brick from this wall are shown in Figure 14. In the vicinity of the spalling cracks there appears an abrupt change in

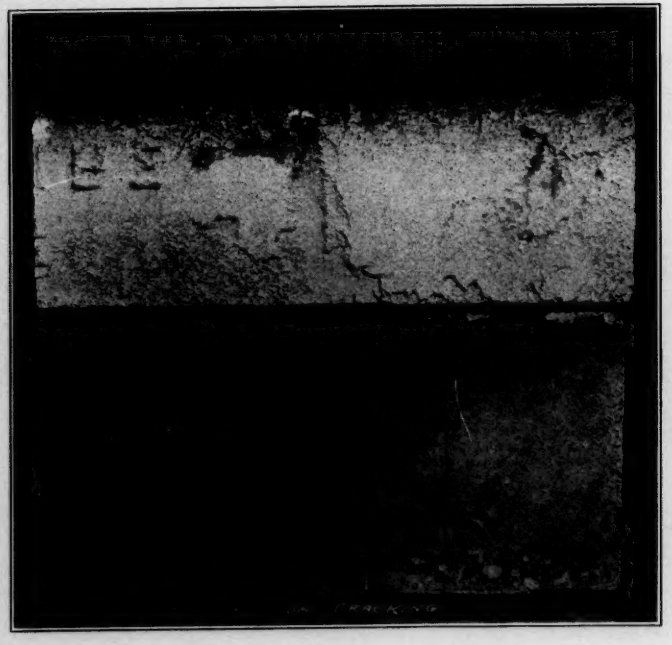


FIG. 4.

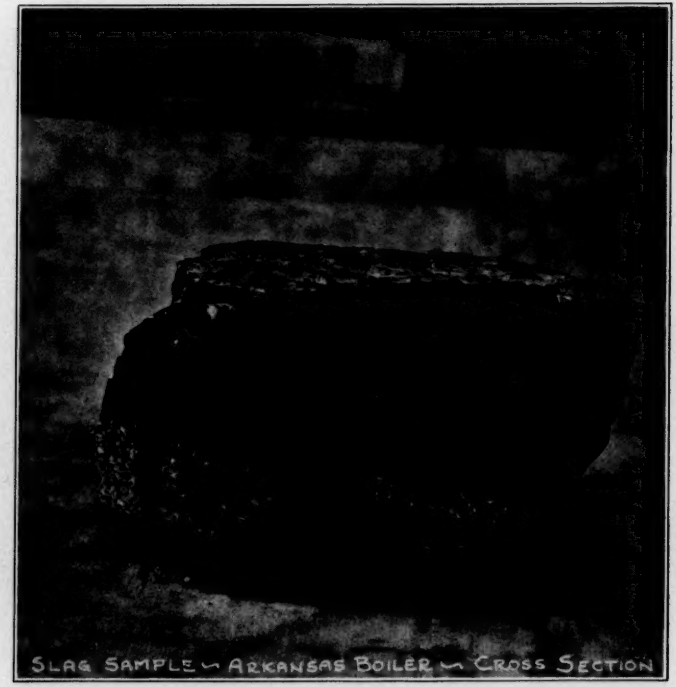


FIG. 6.

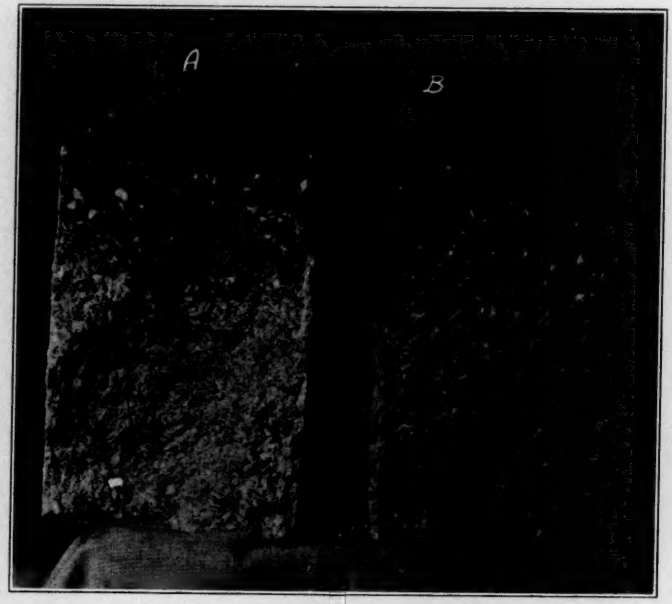


FIG. 9.

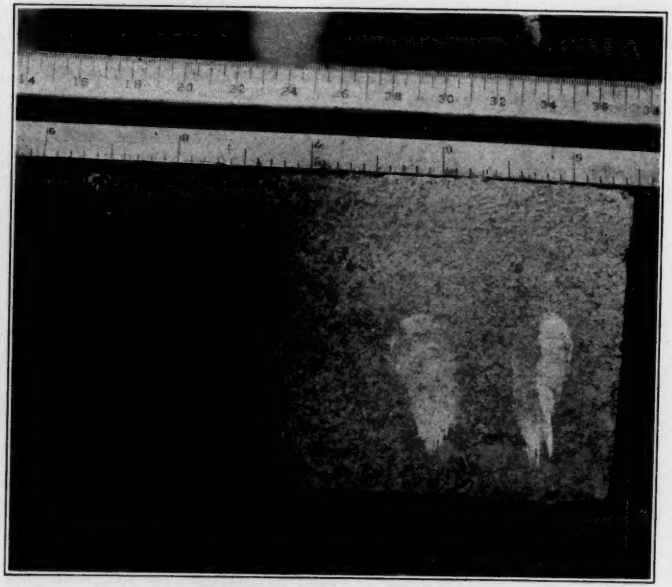


FIG. 10.

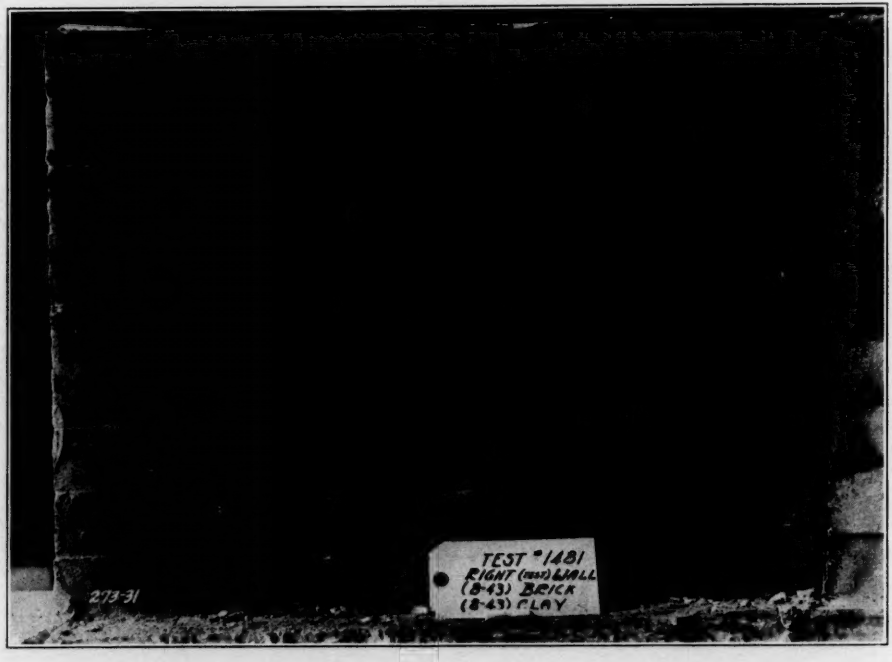


FIG. 11.

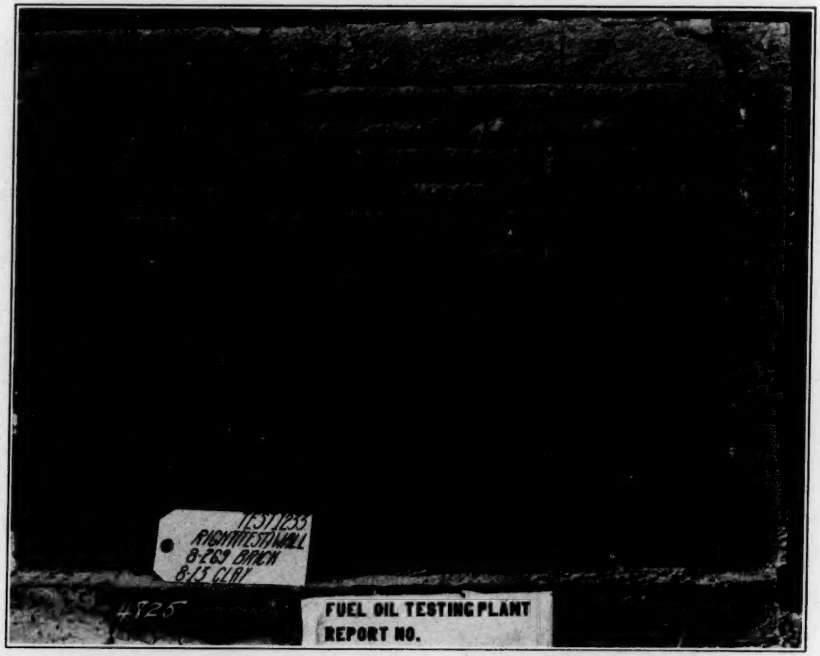

1.51.123
RIGHT TEST WALL
8-2.5 BRICK
8-1.5 CLAY
4525
FUEL OIL TESTING PLANT
REPORT NO.

FIG. 12.

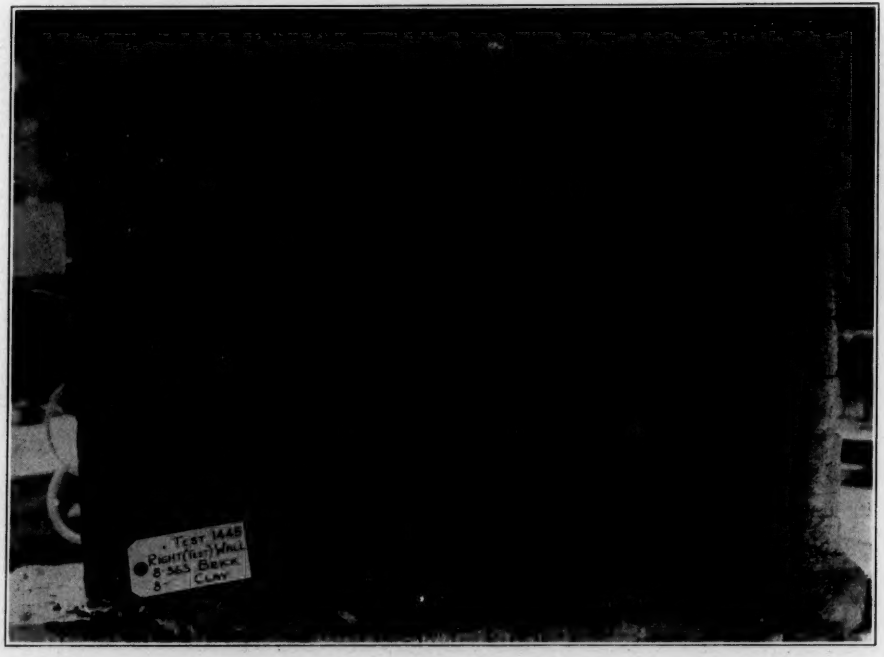


FIG. 13.

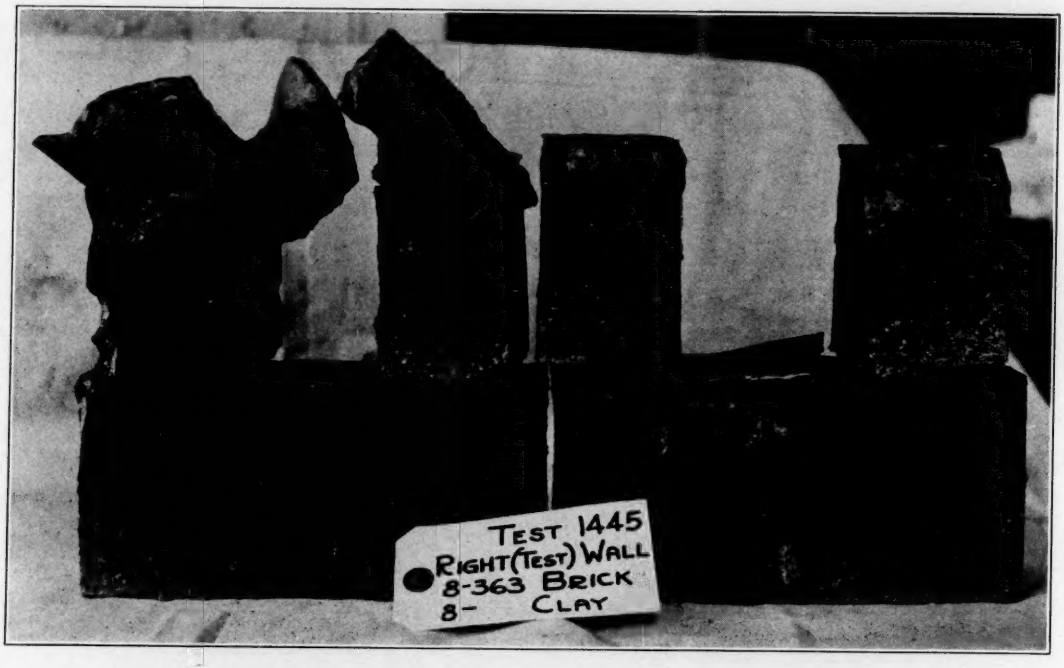


FIG. 14.

FIG. 15A.

FIG. 15B.

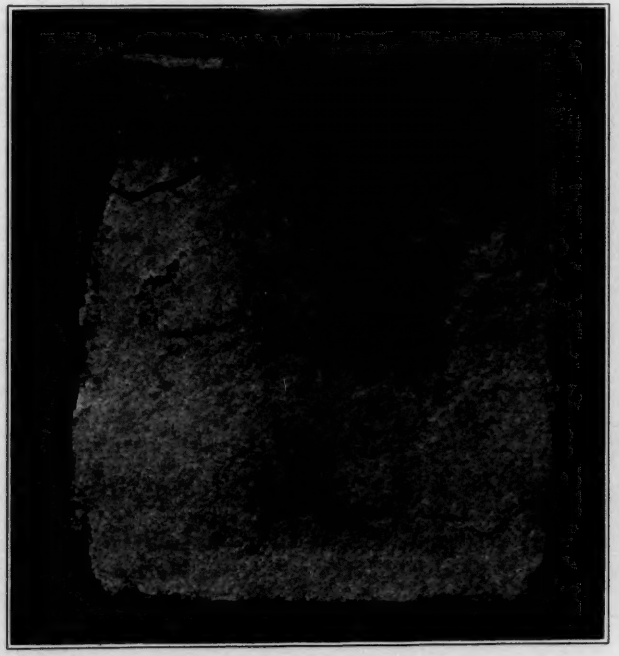


FIG. 15C.

FIG. 15D.

structure. Typical spalling fractures are also shown in Figures 15 A, B, C and D.

Study of the characteristics of brick, the conditions to which they are subjected in operation and the type of failure that occurs should be of material assistance to the operating engineer in obtaining increased refractory life and in determining what conditions influence most the failure of refractories. No matter how carefully firebricks are tested the ultimate criterion of their characteristics is their performance in service.

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THE TREND IN NAVAL ENGINEERING, NO. 5. ELECTRIC WIRES AND CABLES, PROTECTION AND INSULATION OF.

By C. Huey, Civil Member.*

1. The subject of insulated wires and cables aboard U. S. Naval ships, and for marine installation in general, is one that lacks spectacular aspects, but nevertheless, it is evident, that with increasing application of electric power, the reliability of the electric machinery, electric mechanisms and electric lighting, is greatly dependent on the reliability of the insulated wires and cables.

2. To just what extent "reliability" has been fabricated into the completed cables, is best described by separating the component parts into two general classes of materials, (1) mechanical protection and, (2) electrical insulation, and then defining each item in terms of its purpose.

3. *Mechanical Protection.*

(*a*) *Stranding* — All Navy standard cables, except telephone cables, use concentrically stranded copper wires in groups of 7, 19, 37, 61, 91, and 127, and in individual wire diameters 0.010 to 0.091 inch, assembled in combination to obtain the various cable sizes from 4000 to 1,500,000 circular mils. The purpose of this stranding is to give a certain amount of flexibility required for installation, and to obtain the reliability of multiple conducting paths as compared with single wires generally used in commercial work. On telephone cables, the necessity for reducing the mechanical dimensions resulted in the selection of one wire per conductor.

(*b*) *Tinning* — All Navy standard individual wires are tinned, except for portable cables where the tinning interferes with the desired flexibility, and except for telephone cables where *enamel* displaces the tinning. The purpose of the tinning is primarily to protect the copper from corrosion due to free sulphur existing in the rubber or rubber insulation or rubber impregnated fabrics.

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The secondary purpose is to obtain a ready applicable surface for splicing or applying soldered lugs and terminals. It is for this latter purpose that tinned wires are also used on cables which are insulated with materials other than those in which free sulphur exists. It is quite difficult to properly "tin," stranded wires, for applying terminals, if not previously done in fabrication. In the earlier cables, 98 per cent pure unvulcanized rubber, was used as the medium to prevent corrosion.

(*c*) *Cotton Thread* — This is used as a spiral wrapping around assembled stranded multiple conductors, to prevent broken individual small wires from puncturing the enveloping insulation, thereby causing short circuits to adjoining conductors, or to ground.

(*d*) *Jute Fillers* — These are used, as required, to fill the interstices between adjacent insulated conductors of duplex, triplex or multiple conductor cables, to obtain circularity of cross section. This circularity is essential for obtaining water-tightness in stuffing tubes of compartment bulkheads, and terminal boxes. Moisture excluding impregnation is used with the jute, for varnished cambric insulated cables. For rubber insulated cables, dry jute is used, for the reason that the only material available for impregnation tends to soften the rubber in contact. Impregnated asbestos fillers are used on flame-proof asbestos insulated cables.

(*e*) *Rubber-filled Tape* — This material is wound hellically around the cable and is used generally on all fixed cables. It serves as a combination binder, separator and buffer, between layers of insulation, or between the insulation and the armoring or finishing covering.

(*f*) *Cotton Braids* (Inner) in the form of sleeving, is used as a binder over rubber insulation, to reinforce deteriorated rubber, to keep individual conductors separated and to serve as carriers of tracer markings on multiple conductor cables.

(*g*) *Cotton Braids* (Outer) in combination with a weather-proof compound, is used as a finishing covering over unarmored cables.

(*h*) *Reinforced Rubber Sheath* is used on all varnished cambric insulated cables, primarily as a water-proof mechanical protection. It is not used on rubber insulated cables, except on multiple conductor interior communication cables, for the reason that the rub-

ber is inherently a water-proof jacket, whereas the varnished cambric layers are inherently nonwater-proof at the lapped joints. The reinforced rubber sheath consists of rubber-filled, rubber-coated tape, or rubber-filled tape in combination with rubber tape, wound hellically to specified thicknesses and vulcanized into inseparable layers.

(*i*) *Lead Sheath* is used to hermetically seal the inner layers of insulating materials. Due to its inherent weight and lack of flexibility, it has been generally replaced by the reinforced rubber jacket for varnished cambric insulated cables. For rubber insulated cables, its use is not required, as the rubber insulation serves the same fundamental purpose. For special cables, as for radio use or sound motion picture photo cell circuits, the lead sheath is used as a static voltage shield from external high frequency radio effects.

(*j*) *Metal Armoring* (*Basket Weave*) used in the form of soft annealed steel wire, galvanized and painted to reduce corrosion, is applied as on outer covering on varnished cambric or rubber insulated cables, to protect the cables from mechanical injury principally during installation.

(*k*) *Tough Rubber Jacket* — This material in the form of extra high (60 per cent) pure rubber content, of high tensile strength, and molded under pressure to obtain a dense structure resistant to abrasion similar to automobile tire treads, is used as the *outside* covering, for cables used with portable lights, portable tools, portable telephones, welding cables, gyro compass repeaters, and in general, for all movable or portable cables.

4. *Electrical Insulation.*

Rubber Insulation consisting of a vulcanized homogeneous jacket of not less than 40 per cent pure rubber, tensile strength of 1500 pounds per square inch minimum, is used as the principal insulation, only on a limited number of cables, such as the smallest size power and lighting cables which are impracticable sizes for varnished cambric insulation, on cables for radio use, and for interior communication multiple conductor cables which have small size individual conductors. Although quality vulcanized rubber has excellent mechanical and electrical properties when new, due to its rapid mechanical deterioration followed by electrical leakage

in the presence of moisture, it is being displaced wherever practicable by varnished cambric, asbestos, or enamel. The useful life of rubber insulation is about 5 to 8 years in storage, 2 to 5 years in normal service (loaded cable temperature limit 50 degrees C. — 122 degrees F.), and 3 to 6 months in high heat service such as experienced in boiler rooms, steam machinery spaces where the ambient temperature is normally higher than 50 degrees C. (122 degrees F.).

Varnished Cambric Insulation consisting of layers of closely woven cotton material, with three or more coats of a pliant slow-drying baking varnish on each side, is used as the principal insulation on all power and lighting cables, except the smallest size (4000 centimeters) for which it is impracticable. The varnished cambric is applied in the form of a tape 0.005 inch to 0.013 inch thick, in 4 layers minimum, as an insulating wall. The minimum tensile strength requirement of the tape is 2000 pounds per square inch. The dielectric requirements are 800 volts per mil at 25 degrees C., and 300 volts per mil at 75 degrees C. (167 degrees F.). The useful life of varnished cambric insulation in storage is practically indefinite. In normal service, excluding compartments with normally very high ambient temperatures, the life is estimated at from 15 to 25 years. In general, varnished cambric as a class, has been found to have very stable characteristics. Its limitation is based on its characteristics, in common with all vegetable products, of gradual reduction in dielectric strength due to carbonization under long continued moderately high temperature (105 degrees C. — 221 degrees F.), or under very high temperature for a short time (160 degrees C. — 320 degrees F.). The foregoing temperatures are in terms of the actual cable working temperatures or the resultants of ambient temperatures plus heat produced by flow of current. The working temperature limit of varnished cambric for long service life, is generally accepted as 80 degrees C. (176 degrees F.).

Asbestos Insulation consisting of long fiber asbestos, fabricated with not more than 15 per cent of cotton, and impregnated with an inert moisture-resisting insulating compound compressed around the conductors, is used as the inner insulation of power and lighting cables for high temperature installation such as for boiler

rooms, steam machinery spaces, etc. Because of the inherent low dielectric strength of asbestos products, the insulation is reinforced with a minimum of 3 layers of varnished cambric. Heat protection to the varnished cambric layers is accomplished by a further layer of felted asbestos, impregnated with a flame-proof insulating cement.

Enamel Insulation consisting of a flexible, nonhygroscopic compound, chemically inert to ordinary reagents, insoluble in mineral oils, and adhering strongly to the conducting wire, is used as the principal insulation on the 2 to 120 individual wires of telephone cables. This material is used for the reason that it occupies the least space of all the various insulating materials, lends iself to closely twisted pairs essential to telephone transmission, and results in a minimum size cable. The enamel is protected from injury by a double wrapping of silk, and a double wrapping of cotton threads, which serve also as separators, buffers and additional insulation.

5. The foregoing items of mechanical protection and electrical insulation, represent the materials which have been used or are now being used for the various cables on Naval ships. It is of interest to note that the materials listed as mechanical, where the material is not of metal, such as cotton thread, rubber-filled tape, reinforced rubber or tough rubber jacket, serve a dual purpose in that the materials are also either water-proof insulation, or good dielectric materials when dry. The metal "mechanical" items, such as lead sheath, or braided steel, serve a single purpose only as was formally the case when brass or steel conduit was used for mechanical protection.

6. Therefore, the general trend in the design of protected and insulated cables for shipboard use, is to use *dual* purpose materials, with long-life properties under operating temperatures. The materials which are disqualified for dual purpose as mechanical protection are, brass or steel conduit, lead sheath, and basket steel metal armoring. The material which is disqualified as not having long-life insulating properties, is plain vulcanized rubber.

7. The trend for the *protection* phase of the design, is to eliminate the foregoing metal materials, and to use either reenforced rubber or tough rubber, jackets or sheaths. The trend for the

insulation phase of the design, is to eliminate vulcanized rubber for the basic insulation, and to use varnished cambric alone, or in conjunction with enamel, silk and cotton, or in conjunction with water-proofed asbestos for high ambient temperature installations.

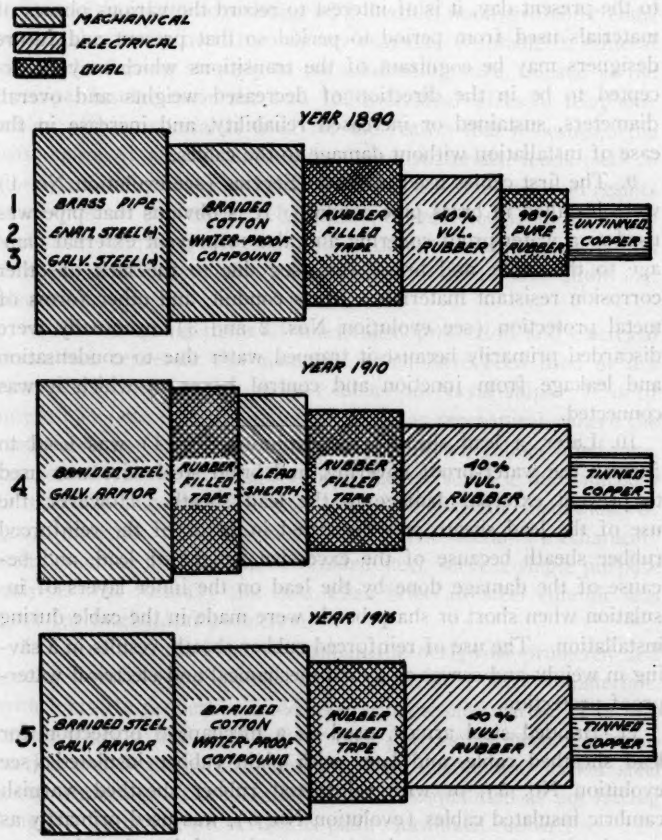
8. In tracing the evolution of protected and insulated cables used in the U. S. Navy, from its original use, about the year 1890, to the present day, it is of interest to record the various phases of materials used from period to period so that present and future designers may be cognizant of the transitions which may be accepted to be in the direction of decreased weights and overall diameters, sustained or increased reliability, and increase in the ease of installation without damage to the cables.

9. The first cables used in Naval Service (see evolution No. 1) were installed in brass pipe conduit. It is obvious that pipe was used as a water-proof covering and also to prevent external damage to the cable, and brass was used due to the lack of other corrosion resistant materials. Brass conduit, and other forms of metal protection (see evolution Nos. 2 and 3), eventually were discarded primarily because it trapped water due to condensation and leakage from junction and control boxes to which it was connected.

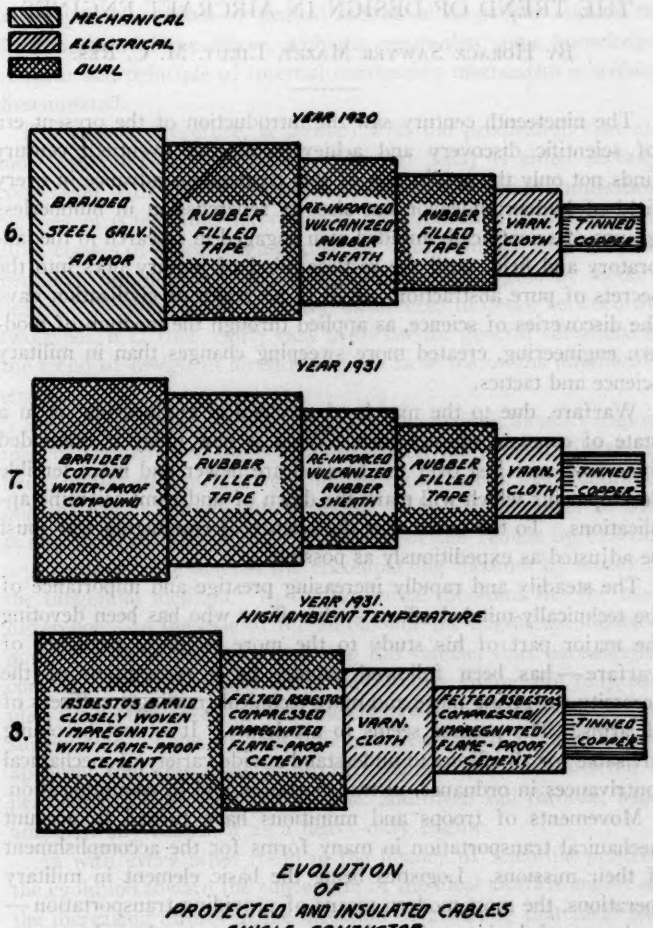
10. Later, a lead sheath (see evolution No. 4) was used to furnish the water-proof covering, and braid steel wire was used to prevent external damage to the lead sheath. However, the use of the lead sheath was discarded in favor of the reinforced rubber sheath because of the excessive weight of lead, and because of the damage done by the lead on the inner layers of insulation when short or sharp bends were made in the cable during installation. The use of reinforced rubber sheath, results in a saving in weight and serves as a dual mechanical and electrical water-proof protection.

11. Braided steel armor, used as a mechanical protection for lead sheathed cable, and later with plain rubber insulated (see evolution No. 5), or with reinforced rubber sheathed, varnish cambric insulated cables (evolution No. 6), was used primarily as a mechanical protection to the outer covering of the otherwise completed cables, principally during installation only. At present the trend is to omit this protection as unnecessary, and to rely entirely

on the protection afforded by the *dual* purpose coverings such as the reinforced rubber sheath, the rubber filled taped and the external water-proofed cotton braid shown as evolution No. 7, for *normal* ambient temperature installations, and shown as evolution No. 8 using asbestos in combination with varnished cloth, for *high* ambient temperature installations.



EVOLUTION
OF
PROTECTED AND INSULATED CABLES
SINGLE CONDUCTOR



EVOLUTION
OF
PROTECTED AND INSULATED CABLES
SINGLE CONDUCTOR

THE TREND OF DESIGN IN AIRCRAFT ENGINES.

By Horace Sawyer Mazet, Lieut. M. C. Res.

The nineteenth century saw the introduction of the present era of scientific discovery and achievement; the twentieth century finds not only the application of this tremendous advance in every field of human endeavor but also its repercussion in numberless groups of scientifically-minded men engaged in research in the laboratory and field, until the restless mind of inquiry prys into the secrets of pure abstraction. And in no realm of civilization have the discoveries of science, as applied through the medium of modern engineering, created more sweeping changes than in military science and tactics.

Warfare, due to the march of mechanical engineering, is in a state of constant obsolescence. What is new today is outmoded tomorrow, and just over the horizon are newer and more terrible developments which will mark the dawn of undreamt scientific applications. To the new order the science of conducting war must be adjusted as expeditiously as possible.

The steadily and rapidly increasing prestige and importance of the technically-minded officer — the officer who has been devoting the major part of his study to the more technical branches of warfare — has been followed rapidly by a recognition of the necessity for giving more and more technical training to officers of all arms. The process seems to accelerate. It is fast becoming advisable for all officers to understand a wide variety of mechanical contrivances in ordnance, in transportation and in communication.

Movements of troops and munitions have turned to account mechanical transportation in many forms for the accomplishment of their missions. Logistics being the basic element in military operations, the most modern means of providing transportation — by the use of the internal combustion engine — is adapted to every need, with the result that today few officers, if any, lack a lively practical interest in such engines. In actual warfare any officer is likely to find himself suddenly confronted with responsibility for

or dependence upon ships, motor trucks in their several forms, tractors, tanks, mechanised artillery, electrical generating sets, pumps, automobiles, motor boats, airplanes or airships. This contingency is recognized by tactical schools in its proper relation to the education of the officer, with the result that some knowledge of form and principle of internal combustion mechanism is widely disseminated.

This article, therefore, is written with the predication that almost every military and naval officer of today is either already conversant with the present trend of design in internal combustion engines, or is becoming interested in the application of this subject. Both classes of officers in any case are quite familiar with the conventional liquid-cooled commercial automotive engine of modern design. Without attention to the differences in detail between the two types, it is, then, fairly easy to follow the general variations in the trend of design in aircraft engines in terms of the automotive engine.

Since both types are internal combustion engines, aircraft engines may be compared to those of automobiles, the general differences being lighter structural weight, higher compression, load conditions and the use of better grade materials. The motive power of both is employed in overcoming the friction of the vehicle by exerting a propelling force against the encompassing medium by means of the tangential medium. In the tangential mediums lies the main difference. In the case of the automobile the engine accomplishes nothing with respect to supporting the vehicle, thus necessitating comparatively light duty, while the aircraft engine not only supplies the motive force but also the component of force required for sustentation. (Not so with lighter-than-air craft.) It is at once apparent that this leads to aircraft engine operation at from 50 per cent to 75 per cent full throttle, and often full throttle, which automatically classes it as a heavy-duty engine.

As with every other trend in the history of scientific progress, the evolution toward the conception of the ideal aircraft engine and the increasing efforts in the direction of attaining that ideal were affected by economic considerations. The extremely rapid progress of aircraft engine design during the last decade is probably due in large part to the fact that the inherent difficulty of the design

problem in these engines has attracted some of the best and most energetic minds to this field, which movement was materially aided by favorable governmental attitudes. The military importance of the airplane in itself and the essential importance of the power plant as a part of the complete airplane, coupled with a general feeling of dissatisfaction with the existing state of aircraft engines, led to the allocation in several countries of liberal government appropriations for research in aeronautical engine design and the solving of various germane problems. This basically economic consideration, and other factors, resulted in the spurt which has placed the present status of aircraft engine design in a position somewhat in advance of any other branch of internal combustion engine design. Not to mention relative achievement.

For many years before and after the notable Wright machine, the first consideration of designers of aircraft engines was the pounds per horsepower ratio. It was for decades agreed that if the total specific weight of the internal combustion engine could be reduced sufficiently, mechanical, man-carrying flight was possible — which contention was, of course, happily substantiated.

Recent achievements of the best aircraft engines in the direction of reliability, lightweight and compactness are remarkable when compared with the slow progress of the early days. Yet the means through which these tremendous advances have been achieved are far from being spectacular. They consist largely in the intelligent application of recent important advances in the science and art of metallurgy and in improvements in detail of design of parts of the engine.

The proven and recognized leaders among aircraft engines of today are all rather conventional in their basic design. They are all carburetor engines; that is to say, they all use a highly volatile fuel, which is first mixed with the correct proportion of air in a carburetor or similar metering device, and then drawn or pumped into the working cylinders. They all operate on the four-stroke Otto cycle. They are distinguished from other types of internal combustion engines chiefly by their systematic elimination of all sources of unnecessary weight.

In this systematic weight reduction modern engines of this classification first of all seek to employ light alloys of aluminum

and/or magnesium wherever possible. In parts of the engine where the nature or magnitude of the stresses (as in the crankshaft), the temperature conditions (as in valves and valve seats) or other special conditions forbid the use of the light alloys, the designer uses the finest kind of steel—usually an alloy steel—selecting it for a particular formula and heat-treatment which exactly fits it to the functions of the part being designed.

After taking the greatest pains to select the best material obtainable for each part of the engine (a selection which would be ruled out on grounds of excessive expense in designing commercial automotive engines) the designer proceeds to concentrate tireless ingenuity upon the design of each part in order to use the minimum quantity, by weight, of the chosen material. Although giving some consideration to the cost of machining the part under production conditions, his first thought is for the elimination of weight without sacrifice of that essential minimum of strength which spells both reliability and reasonable durability. Here again he departs a long way from the standards of the designer of commercial automotive engines, in calling for parts to be machined all over, ground, polished, etched with acid for metallurgical reexamination, subjected to X-ray or magnetic analysis for hidden flaws, and filed, scraped or polished until it comes within certain narrow limits set for its weight.

Integral features of internal combustion engines vary in design importance with the conception of the function for which they are built. The most desirable features sought by all designers of aircraft engines, in order of importance, are:

Reliability. This quality carries most weight because of the fact that the airship becomes helpless and the airplane must descend to earth if the power plant fails. It is the engine's ability to continue running and developing the required power for a reasonable period (the normal continuous-service period for the particular service in which the engine is used — often corresponding to the capacity of the fuel tank) or continuous power output in terms of propeller thrust, without being shut down for any repair or readjustment.

Direct Operating Economy is worthy of considerable attention because of its effect on the cruising range of the individual air-

craft. It is considered to be the minimum practicable consumption of fuel and lubricant per horsepower per hour to maintain continuous operation under normal conditions.

Lightness, in that it directly affects take-off, climb and speed as well as cruising range, receives emphasis from a designer. Low specific weight, usually expressed in terms of the pounds per horsepower ratio, is desirable since it is closely allied with direct operating economy.

Smoothness. Absence of vibration is essential even in the crudest applications because of its influence on the reliability and durability of the engine, its relation to the structural characteristics and structural weight of the machine, the usefulness of the instruments, the effectiveness of the crew, fatigue of the pilot, comfort of the passengers, etc.

Compactness receives attention because it reduces parasite resistance or power plant drag factor affecting direct operating economy by its effect on maneuverability, and degree of vision permitted the pilot. A high increase in efficiency of the complete airplane can be obtained through the use of a more compact engine. As for the term itself, compactness is moving very definitely away from the old conception of square feet projected frontal area toward separation into (1) Aerodynamical drag, definitely measured or accurately calculated, and (2) Interference with the pilot's vision expressed in terms of percentage of a sphere of perfect vision in all directions.

Ease of Maintenance. This is only one of many examples of the present day movement toward a thorough practical analysis of engine design problems, toward removing obscurity from our concepts and ambiguity from our definitions. With compactness it has come to receive increasing attention from the designer, and is just beginning to receive very effective emphasis in the more standardized designs of engines intended for quantity production. Factors which bear on this feature are fewer parts which require care through the self-oiling and like improvements, many parts which receive less frequent attention due to the use of new metals in valves, bearings, etc., greater accessibility of parts for inspection, quick-detachable cowling, rocker-box covers, etc., location of oil strainer or relief valves and other improvements in location of functional parts.

Ultimate Overall Economy which incorporates the depreciation factor on innate durability and first cost, on insurance rates or risk borne by the operator, and finally by repair costs on labor and materials for replacements. Durability remains a quality not directly sought after but nevertheless achieved in considerable measure as a collateral of reliability. At present, due to advances in research, engines become obsolete in design before they wear out. All these factors directly affect long term operation cost, which is sometimes a contributing factor in a sale, particularly in commercial aviation.

Salient characteristics in the trend of design in aircraft engines are (1) Form — chiefly exemplified by cylinder arrangement, (2) Improved Materials, (3) Improved Detail Design, (4) Cooling, and (5) Fuels.

For some years before the accomplishment of man flight, internal combustion engines were regarded as the ideal prime mover for heavier-than-air craft. Designers and manufacturers (what few there were) believed that in the reduction of engine weight lay the bare possibility of mechanical flight, and of course actual human flight was but a step further. Thus the Wright Brothers, spurred on by the success of their motorless gliders, set themselves to the task of constructing an engine lighter in weight per horsepower than any which had ever been built.

Their first attempt at flight was made with an engine of 12 horsepower at 1200 revolutions per minute, weighing 170 pounds dry, or about 14 pounds per horsepower. The inventors actually achieved flight with their second engine (Type B) of 35 horsepower at 1200 revolutions per minute. It was a vertical, water-cooled 4 cylinder type with a bore of 4.375 inches and a stroke of 4 inches, weighing 180 pounds. Individual cast iron cylinders were jacketed around the barrels only with aluminum jackets and no provision was made for cylinder head cooling. Interchangeable valves stood vertically in cylinder heads, the inlets operating automatically and the exhausts by push rods and rockers from a cam shaft contained inside the crankcase. A flywheel was attached directly to the crankshaft, as were two sprockets from which chains drove two pusher propellers at a .272 ratio. Singularly enough, the engine had no carburetor, but a small gear pump discharged fuel

into the spread end of the inlet pipe through a small jet orifice, which regulated the speed of the engine. A pump geared to the camshaft actuated the force-feed lubrication system.

Thus in these first engines lightness was attained by the application of light materials and new design to the conventional 4 cylinder in-line automotive type, and we shall see how, since 1903, this elementary type has made its periodic reappearance among aircraft engines as various shapes lost favor through the introduction of improved materials and improved detail design time and again.

At about this same period, in fact just a bit previously, a much more original project was being carried out by the scientist, Prof. S. Pierpont Langley, and Mr. C. M. Manly, who were seeking to substitute a gasoline engine for the Langley steam engine with which his aerodynamically-correct "Aerodrome" flying machine was powered. While Langley perfected his airplane models Manly constructed an internal combustion engine, air-cooled, radial, with 5 cylinders of 2 1/16 inches bore and 2 3/4 inches stroke, developing 3.2 horsepower at 1800 revolutions per minute for a short period. Its weight with carburetor, ignition and small storage battery was only 10 pounds. With this engine Langley's quarter-size model made a steady, short flight in August, 1903, four months before the Wright Brothers' epochal achievement.

The full-size Manly engine never participated in aerial flight, but it is now universally recognized that nothing but faults in the airplane launching gear prevented this remarkable engine from gaining the distinction of being the first to make possible man-carrying mechanical flight. The illustration shows a cross-section of this engine, which was also a five-cylinder radial, but water-cooled. Its bore was 5 inches and the stroke 5 1/2 inches, developing 52 horsepower at 950 revolutions per minute for a dry weight of 125 pounds. The engine completed a dynamometer test of 30 hours some time later at full power in excellent condition. *With all accessories the weight totalled 3.6 pounds per horsepower* which is not a bad ratio today! For 1902 this performance was unbelievable; twenty years were to pass before anything distinctly better than the Manly engine was designed, and nearly 10 years before its essential features were resurrected from limbo and suc-

cessfully incorporated in radial engines such as the Anzani, the Canton-Unne (later the Salmson) and in numerous war-time rotary engines. Conventional only in using water-cooling and the four-stroke cycle as a five-cylinder single crank radial embodying extremely light weight design in all its parts, it was one of the most radical departures in the whole history of aircraft engine design. It was an anachronism, twenty years old the day it was born, and as a singularly ironic paradox this engine which was never to fly was the most significant aircraft power plant ever to be designed. But contemporary builders and others to follow were off on other tangents which diverted attention for two decades, and Langley and Manly were forced into obscurity through the failure of their flying machine launching apparatus.

Manly's engine was not only a milestone in engine design, but it represented about the first and last time an innovation in shape or form brought any real advance in such engine. What is claimed to be or thought to be new and radical with engines which make a meteor-like appearance in a world quick to grant recognition to any sound labor-saving idea, has its counterpart in the history of design long since forgotten. Features that are commonly considered to have originated since the war in many of our present type production motors, in reality embody no new principle or invention; their prototypes were experimented with when successful aviation engines could be named by every callow enthusiast or airdrome satellite.

All the most successful aircraft engines in production today are quite conventional in all their features, utilizing shapes and cylinder arrangements which were used in the earliest days of flying and whose origins are, in the majority of cases, lost in the present day plethora of publicity. Brig. Gen. William E. Gilmore in his "The Development of Air-Cooled Engines" (Airway Age, Volume 9; No. 7) says, "It has seemed so eminently fitting that an air-cooled engine be used for the propulsion of aircraft, that no one has been particularly surprised at its recent really amazing advance. Those who have been least surprised are those who have forgotten that the air-cooled engine is practically as old in air activities as the water-cooled, and that in the early days of fly-

ing it played an extremely important part. Most of us have forgotten the pioneer air-cooled engines, the Renaults of 1907, the Anzanis of 1908 (with which Bleriot first crossed the Channel via air), the Adams-Farwell rotary, and the early Curtiss air-cooled engines. We have nearly forgotten the Clergets, the Gnomes, the Le Rhones, the A. B. C.'s, the Br-2's and the R A F engines of the war. The period of apparent inactivity in air-cooled engine development immediately after the war . . . has erected a barrier our memories cannot easily surmount."

Aircraft engines are invariably divided into two main classes: (1) rotary, and (2) static or stationary, either water- or air-cooled. Engines of the stationary type may be divided further into the horizontal, Vee, vertical, radial, W, X, and Y shapes. Since the inception of aircraft engines, devious designs, shapes and types have been evolved, but at the present time the above types include all existing aircraft engines, and those of the preceding paragraph are representative of the two main classes.

Following the primitive Wright engine there were any number of engines with their cylinders arranged in tandem, *i.e.*, one behind the other. For a period of years this type, doubtless influenced by the prevailing design in automotive engines, maintained unquestioned supremacy over other designs, in this country at least, and it was *not* the best adapted to air-cooling. Airplanes flown with so small an amount of available power were of large wing area, slow in speed, and sluggish in movement, and even the most daring pilot could not coax more than six or seven hundred feet altitude from his stick and wire contraption. An infinitesimal power safety factor was in part responsible for the few birdmen who would venture into the air, with elaborate hocus-pocus of preparation, if the tops of the grass were swaying in a seven-mile-an-hour gale, and the most glamorous pilots of that day were intimidated unless a handkerchief suspended from the hand hung limply without motion from the wind. To disregard these criteria was to court the title of supreme dare-devil of all. More power was the cure.

Closely following this first Wright model came the 6 cylinder vertical and 8 cylinder Vee engines of 65 and 75 horsepower. All these were too light and frail for sustained service, and flights

therefore were reckoned in minutes, not hours. High pressure lubrication to vital parts was not yet developed, and valves warped with disheartening regularity. Little time in the life of a plane was spent in the air; overhaul and repair were the daily fare of mechanic, airman and owner.

Model E-4 was the first Curtiss water-cooled engine, a 4 cylinder vertical rated 50 horsepower at 1500 revolutions per minute and weighing 5 pounds per horsepower. The cylinders were made of cast iron and fitted with copper water jackets; the inlet valve was located on the side of the cylinder and the exhaust valve on the cylinder head, while battery ignition and splash feed lubrication were employed. This was preceded by the model B-8 air-cooled adaptation, having a bore of 3.625 inches and a stroke of 3.24 inches, rated 40 horsepower at 1800 revolutions per minute. It was the first Curtiss engine to be used in any aircraft, but it was copied after the conventional motorcycle engine of that day.

In 1909 the Curtiss model L, in a Curtiss biplane, won the Rheims Race, placing it among the leading aircraft engines of the time. It produced about 50 horsepower, and weighed 250 pounds dry, being an 8 cylinder Vee. Other makes of air engines which were prominent in the past were the 40 horsepower Hall-Scott 4 cylinder vertical water-cooled, the Kirkham 1911 6 cylinder vertical water-cooled, the Renault 8 cylinder air-cooled Vee which won the Michelin Cup in 1910 by a non-stop flight of nearly eight hours, the German Mercedes 6 cylinder vertical water-cooled of 100 horsepower at a ratio of 4.5 pounds per horsepower, the British Green 4 cylinder vertical water-cooled of 35 horsepower, which won several important contests, the Anzani radial, and others which enjoyed a passing popularity. Now that weight had successfully been reduced to a point where light engines were common, the other troublesome details were receiving increasing attention from experimenters which resulted in many variations of the original Vee and vertical cylinder arrangements.

Later on, following the practice in motorcycle engines, there were countless experimental models which arranged two or more cylinders around a single crankpin, thus producing horizontal opposed 2 cylinder engines, Vee twins and fan types of 3 and 5

cylinders. The most successful precedent of the Manly engine was not followed for some years, in considerable degree influenced by the failure of Langley's "Aerodrome" at launching.

It required no prodigious brain to see that a combination of the tandem idea and the fan-shaped idea in engine design might produce a highly desirable type from many points of view. This short step was taken before long, and resulted in a Vee type engine with two rows of cylinders in tandem, each crankpin having two pistons working on it.

The next departure was, of course, the W type engine with tandem cylinders and three pistons working on each crankpin. One of the first combinations of the tandem and fan systems was the 40-50 horsepower Renault (France) in which the 8 cylinders were attached to the crankcase by four long bolts and a cruciform yoke, similar to early motorcycle engine practice. A notable feature about this machine was that the cylinders were cooled by an air stream from a centrifugal blower type fan mounted on the end of the crankshaft. It was a serviceable design. Another engine with the same basic form was the Curtiss L water-cooled 8 cylinder Vee, also mentioned above.

Types employing the fan shape were Anzani, Canton-Unne and R. E-P (France). One of the earliest Anzanis propelled Louis Bleriot and his monoplane across the English Channel in 1909, which flight shook the aeronautical world, and the rest of the world too, with the impending possibilities of aviation. This power plant was a 3 cylinder, single crankpin air-cooled engine of 24 horsepower at 1600 revolutions per minute, weighing 145 pounds. Cylinders and head were integrally cast. Uneven firing resulted from the fact that intervals of 120 then 300 degrees elapsed between firing strokes, for an engine whose cylinders were set 60 degrees apart. This defect soon led to the construction of a double fan type, with 6 cylinders and a double throw crankshaft, but it was found necessary to cool the rear cylinders with water, so the entire engine was made water-cooled. Then came the single-throw radial with intervals of 120 degrees between cylinders for smoothness, and this engine, especially in the rotary adaptation, was adequately cooled by air. This marks another milestone in the development of aircraft engines.

The distinction existing between fan type and multiple-tandem types (such as the V, W, X, etc.), is a distinction in degree only. Multiple-row fan types *tend* to use a staggered arrangement and to use less than four rows of cylinders. Multiple-tandem types *tend* to use four or more cylinders in each row, but the multiple-tandem idea has yet to be fully exploited. Its highest development to date from the point of view of broad basic conception is undoubtedly the Farman (France) W type engine of about 700 horsepower with 18 cylinders.

With such slight differentiations between our necessary classifications, therefore, we find all possible shapes being experimented with. Fiat (Italy) built an 8 cylinder air-cooled Vee which weighed 150 pounds. The entire engine in flight was encased in a cowling arrangement which directed the turbulent slipstream around the cylinders with the desired results. Napier and Son, Ltd., London, built the Napier "Lion," one of the best-known W type engines, which was accepted by the British Air Ministry at the close of the war. It had three rows of 4 cylinders each, and was rated 450 horsepower at 1925 revolutions per minute, the weight being 858 pounds, or 1.9 pounds per horsepower. Lorraine-Dietrich was another manufacturer of the W type, while Hispano-Suiza (France) in a more modern adaptation of this shape, produced a 12 cylinder motor with three banks of 4 cylinders each, weighing even less and delivering its rated 450 horsepower at 2000 revolutions per minute.

During this experimental era there were even engines in the shape of an X, with two or more tandem cylinders in each row. The modern Packard X type water-cooled engine has been one of the best as regards pounds per horsepower. A similar experimental engine built by Viking was a 16 cylinder air-cooled model, reputed to produce 140 horsepower at 1600 revolutions per minute.

Perhaps, in the welter of mechanical anomalies, there was some unsung hero who built an engine with 5 or more rows of cylinders, for evidently no system was too radical to prevent continuous experimentation. There have been models such as the Demont double-acting rotary air-cooled 6 cylinder engine developing a remarkably low weight power ratio of .73 pound per horsepower, the Potez upright vertical air-cooled 4 cylinder model with a

vertical crankshaft and bevel-geared propeller hub, the Salmson horizontal radial engine of proved performance, the Statax air-cooled rotaries in which the axes of the cylinders were arranged parallel to the axis of rotation, the 18 cylinder Tips rotary, the Shubert valveless air-cooled radial, and the Caminez cam-operated 4 cylinder air-cooled radial.

As might have been expected in the manufacture of so many forms of engines in each of which was sought more and more efficient output, not a few empirical and fantastic models were constructed along impractical lines, eventually to be relegated to limbo with the forgotten Manly engine. Unfortunate restrictions also, it must be remembered, were prevalent in metals, controls, mountings, propellers, stress analyses, ignition, spark plugs, lubricants, fuel and a host of other contributory factors which directly affect performance, so that any engine which won a commendable degree of relative success had been through a long and painful travail.

After the efforts of Manly and Wright, improvements continued to be sought mainly in weight–reduction, but the other factors — reliability, direct operating economy, smoothness, compactness, ease of maintenance and ultimate overall economy — began one by one to receive attention from various experimenters and inventors, so that the airplane as we know it today is a product of a vast combination of individual efforts and brains.

The two main lines of development at this particular time were in the water-cooled tandem types and in the air-cooled fan or radial. Indeed, the history of changes in airplane engine design shows that the main differences were in the arrangement of cylinders first and foremost, while adaptations of improvements in materials and specific detail design were corollaries.

Probably because of certain elementary and ill-conceived notions about air-cooling problems, a type of engine which might best be described as the staggered, multi-row radial engine was given much more attention than the radial-tandem type, or a combination of the radial and the tandem principles. One such model was the Salmson 200 horsepower 14 cylinder radial *circa* 1914.

Chief among early developments was that toward providing adequate cylinder cooling, with particular regard to air-cooling.

The added weight of water-jackets, radiator and cooling fluid was a formidable factor in the weight-reduction attempt, and the flanged, air-cooled cylinder seemed to offer great possibilities providing certain definite undesirable features could be eliminated. Difficulties encountered in the cooling researches, due to extremely bad cylinder-head design and weight involved in the flywheels so often retained in those days led to the adoption of the idea that the rotary engine would prove an ideal solution. At just before and during the early days of the war the air-cooled rotary became the most widely used type in practically every country involved, except Germany which developed water-cooled in-line engines to a greater extent without, however, entirely forsaking experimentation and production of the rotary. It will be noticed that Germany's emphasis of the vertical in-line type was a distinct recrudescence of the original aircraft engines.

The cylinders of a rotary or revolving-cylinder engine, designers found, provided adequate cooling drafts by their own rotation, independent of the airplane speed. This was hailed as a big step in favor of the air-cooled principle. So we find at this critical period in aviation history the rotary idea best known in the Le Rhone, Clerget, Gnome, Monosoupape and Bentley (Br-1, Br-2) engines which carried many an Allied airman to victory, and copies of which flew not a few German aviators to success as well.

Yet the temporary ascendancy enjoyed by the rotary engine at this time could not endure. Recognized, distinct limitations were all too evident, even though other engine design principles were neglected momentarily due to faulty air-cooling and the weight of water as well as the factor of radiator vulnerability under fire. The rotary introduced serious engineering complications and compromises. Not only was a large proportion of developed horsepower diverted by the necessity of rotating the cylinders themselves through the air, but the greater the weight of cylinders grouped around one crankpin the greater the difficulty of increasing rotational speed because of enormous centrifugal forces, a problem not found, of course, in the tandem engine.

Rotaries were also undesirable because of excessive lubricant consumption due to centrifugal force. The castor oil used for lubrication in these types possessed superior qualities under the

unusual operating conditions of a centrifugal oiling system in which oil was fed through the stationary crankshaft and into the crankcase, whence it was dispersed out the exhaust valves of the cylinders, literally thrown out into the air. Other considerations which precluded permanent adherence to the rotary principle were the high inertia forces of rotating cylinders and crankcase which must be overcome, the inherent interference of centrifugal force on pushrod and valve operation, and the gyroscopic forces working to the detriment of plane control, not to mention inherent distribution and carburetion difficulties.

Meanwhile, the elements of successful *fixed* radial air-cooled engines existed with Anzani in France, A.B.C. and Cosmos (later Bristol) and Armstrong-Siddeley engines in England, and several others. Lawrance in America about 1916 worked out the same problems with notable results. Simultaneously a great many people were engrossed with the tandem idea in both single and double row form. Volumetric efficiency in terms of horsepower produced per cubic inch piston displacement constantly increased because water-cooling had attained a comparatively higher degree of perfection than formerly, and because in these engines lighter weight of fixed structural parts was given more attention. Corresponding advances made in the design of commercial automobile engines were more often directly applicable to this type than to the others. The tandem engines of this period evolved from cruder types with valves operated by pushrods and using a great deal of cast iron, copper and brass (such as the Curtiss OX-5), to types using overhead camshaft and cleverly built up steel cylinders (as employed in the Liberty and some German engines), to types in which the crankcase was relieved of some stresses by joining all the cylinders with a stiff cylinder-head-and-camshaft housing, usually cast from aluminum.

What appears to be the final development of this excellent improvement (which was pioneered by Hispano-Suiza in France during 1915) is the casting of the entire row of cylinders in one aluminum block capable of taking many strains which must be carried by the crankcase in engines with individual cylinders. The Hispano, with this advance, marked the beginning of the decline of rotary engines in 1916, for far better results were

obtained with one casting for all cylinders than from engines with separate cylinders which could not distribute stresses to other parts of the series. The new Lorraine-Dietrich "Courlis" (1930) for example is another illustration of the tendency to follow the Hispano school of en bloc cylinder design, definitely abandoning the separate arrangement.

In Germany, where water-cooled engines had not lost ground, the Mercedes was by far the best known, during the war being one of the standard makes and very popular with the pilots. Their favorite model was a 160 horsepower 6 cylinder vertical with individual cylinders but with an overhead camshaft and aluminum crankcase. Later, needing more horsepower, their first departure from the revived in-line water-cooled 6 cylinder type was the 500 horsepower 12 cylinder Benz Vee.

England's contribution to the new trend was, at first, the Napier "Lion" which appeared during the last days of the war. It proved to be quite efficient and of admirable properties. The 12 cylinder "Lion" is a W type formed in three rows of four cylinders each, and develops approximately 500 horsepower at 2050 revolutions per minute with a geared propeller. Its weight is 858 pounds, or 1.9 pounds per horsepower.

With both armies the steadily increasing importance of aviation during the world war naturally resulted in the production of numerous aircraft engines by all participating nations. The Rôlls-Royce, Siddeley, Napier, Sunbeam, A. B. C., in England; the Hispano-Suiza, Lorraine, Renault, Salmson, Anzani, Clerget, Gnome, in France; the Mercedes, Benz, Maybach, Daimler, in Germany and Austria; Fiat and Isotta-Fraschini in Italy and others too numerous to mention made their appearance in various models and in rapid succession.

The recognized fact that nations with the fastest and best planes and engines would gain and maintain supremacy in the air rendered obsolescent each new design shortly after its advent at the front, for experimentation was advancing at a furious rate. More power and higher rotational speeds were the cries from pilots, and builders back of the lines responded.

As more and more horsepower and speed were demanded, the power limitations of the rotary, combined with other undesirable

qualities in this type of engine, resulted in its increasing unpopularity. Water-cooled Vees with emphasis on rotational speed and low weight predominated within a relatively short time, and this condition persisted until about five years ago, as we shall see.

At the close of the war the school of design represented by most of the German 6 cylinder in-line engines, the American Liberty, the Packard, Rolls-Royce and some French and Italian engines, had about reached its zenith. Other designs, derived from the basic advances of the Hispano-Suiza, were beginning to gain ground. The Wright Aeronautical Corporation in the United States took the Hispano design complete but improved it rapidly until it was hardly recognizable except by its external shape. The Curtiss-Kirkham and later Curtiss engines, culminating in the D-12, the V-1440 and the V-1570, improved the same basic ideas along still other lines. Other makers felt their way from the individual built-up steel cylinder of the German adaptation through the intermediate type of design in which steel cylinders and jackets are joined by a single cylinder head casting running the full length of the tandem row of cylinders and gaining additional stiffness from the cast aluminum camshaft housing above it. Packard took this step about 1923 in developing its new 1500 cubic inch and 2500 cubic inch engines. Wright in its Tornado "T" series development cast the cylinders in blocks of three, but the final experimental model of the "T" series had blocks of six cylinders.

Meanwhile Curtiss adhered to the block idea and developed some Vee type water-cooled engines which were so good that many of their features were extensively copied.

During the last years of the war and the early post-war period, when the tandem type water-cooled engine seemed to be supreme, a few capable designers, convinced by cool logic and scientific study, stuck to the idea that an engine for air work should be air-cooled, and that a radial form offered greatest possibilities of weight reduction and satisfactory air-cooling. The well-known air-cooling efforts of Charles L. Lawrance in this country began in 1916, during which year three engine designs were laid down and two of them constructed. His 1916 project embracing a 9 cylinder radial air-cooled engine was submitted to the President's

Aircraft Board at the time of our entry into the war. The Board felt, however, that the Liberty and the Hispano-Suiza types would meet all their requirements and that the expense of air-cooled development during war time was not justified. It was this same design, brought up to date through further experimentation, that became the J-1 engine of 1920, forerunner of the famous "Whirlwind."

The first static radials able to stick their heads above the surface of water-cooled current opinion were the air-cooled A. B. C. and Cosmos engines in England; the Wright R-1 and R-2 radials (largely developed through the engineering work of the Army Air Service in the United States) and a little later the Armstrong-Siddeley series in England and the Lawrance engines in America.

These air-cooled static radials gained headway slowly. The British Jupiter engine soon became fairly well known abroad. But before long it was partially eclipsed by the Armstrong-Siddeley and the Lawrance. A merger of the Lawrance interests with the relatively powerful Wright Aeronautical Corporation in the United States gave a tremendous impetus to the air-cooled radial, and between 1923 when the merger occurred and the end of 1926 the Lawrance engine was constantly undergoing revision. The net result was that under the new title of Wright Whirlwind the engine jumped from comparative obscurity to national and then international recognition as one of the most advanced engines of the time. Everywhere this design was creating friends by its long-term, reliable operation between overhauls and its general adaptability and efficiency.

Another make of similar engine which has won high commendation is the Pratt & Whitney radial (U. S.) The smaller model, the "Wasp," and its big brother, the "Hornet," are extremely popular with the Army and Navy aircraft squadrons, being adaptable to service use under extreme conditions. Many other engine companies started production in 1927 and 1928, riding for a time on the crest of the wave of air-cooling popularity, but, of course, since that time there has been a high rate of mortality.

The trend of design during the last few years shows nothing radically new in the form or shape of successful aircraft engines,

although much has been accomplished in improvements of material and specific detailed design. It shows, however, that there has been a tendency to emphasize first one and then another of the desirable characteristics of the ideal engine. It shows evidence to prove that a lamentable lack of the conception of the ideal engine as a whole resulted time after time in roundabout design as a substitute for basic research, leading often to cures as bad as the ills. It shows the adoption of the results of research in allied fields such as (a) metallurgy (b) thermodynamics (c) production engineering of machines, tools and methods in cutting and grinding metals (d) special machine design as exemplified in high-performance gears, superchargers, magnetos, starters and carburetors. And it shows that when engines of maximum practicable displacement of individual cylinders have been developed to their limit of thrust horsepower, engineering and research may point the way toward new types producing even greater thrust horsepower than is now deemed possible, but that such progress is not immediately probable.

The most highly-perfected and efficient proven engines of today in form are single row air-cooled radials (such as P. & W., Wright, Bristol, Armstrong-Siddeley, Warner, Pobjoy, and Lycoming) and liquid-cooled Vee tandem engines (such as Curtiss, Rolls-Royce, Hispano, Fiat, Isotta, etc.) and liquid-cooled W tandem engines (such as Farman, Hispano, Isotta). Practically all of them run at 2000 revolutions per minute or higher. Propeller reduction gearing is generally provided as an optional feature. The best are equipped with some form of supercharger, although the models intended for low altitude, high-duty work may use the supercharger mainly for improved mixture-distribution. Many have all working parts completely enclosed and provided with reliable, built-in continuous lubrication. Cleanness of external appearance is marked in comparison with engines of less successful design. Weight is low, form is compact, fuel consumption is usually below .55 pound per horsepower per hour, oil consumption is generally below .03 pound per horsepower per hour, such few parts as require routine attention are reasonably accessible, and reliability is phenomenal considering the high-duty service

demanded almost incessantly by military and commercial operation.

As before illustrated, the desirable features sought by all designers of aircraft engines are responsible for the trends. It is partially as a result of the recognition of compactness desirability, for example, that the inverted, air-cooled tandem engine has received so much attention in the last two years, and that the radial engine has almost universally adopted the ring cowl to reduce total engine drag, especially since much cannot be expected here in reduction of frontal area—the more fundamental factor in both drag and vision interference. Ease of maintenance, overall operating economy and long life have been responsible for the adoption of some features and the rejection of others throughout the history of aircraft engine manufacture. The increasing tendency to do away with pushrods and substitute overhead camshafts is a case in point.

But directly affecting the whole gamut of changes in design, both large and small, has been the constant improvement in fuel and lubricants. Today high-compression engines are both common and economically practicable, for newer and better fuels and lubricants have made possible the design of more efficient engines year after year. The success of present-day engines is no more remarkable than the scientific research which makes them possible and the increasing recognition of what will constitute the ideal aircraft engine.

The preeminent position of the leading types named above seems today to be threatened from only a few quarters:

(1) Air-cooled tandem types conventional in most respects now being extensively developed (Cirrus Gipsy, Chevrolet, Rover, Menasco tandems—Wright, Allison, Isotta, Vee tandems).

(2) Multi-row radial types, mostly in air-cooled and quite conventional design (Armstrong-Siddeley and several semi-secret American experimental models.)

(3) Fuel-injection engines in all sorts of types and cylinder arrangements (Packard, Junkers, Clerget).

THE NEW DOLLAR LINERS, S.S. *PRESIDENT HOOVER* AND S.S. *PRESIDENT COOLIDGE.*

A Bbief Description of the Vessels, Machinery and Their Trials.

By Henderson B. Gregory, Member.

The Dollar Steamship Lines have recently added to their fleet two twin screw palatial steamers, the *President Hoover* and the *President Coolidge.* They were designed and constructed by the Newport News Shipbuilding and Dry Dock Company at a cost of approximately $8,000,000.00 each, including equipment furnished by the owners, and they are the largest passenger vessels built to date in the United States. They conform to the American Bureau of Shipping Classification ✠A–1 Ⓔ with freeboard and the Navy Department approval as prescribed by the Jones-White Shipping Act of 1928. Both vessels are now on their maiden voyages to the Orient, the *President Hoover* having sailed from New York, August 6th, and the *President Coolidge,* October 15th.

BUILDING RECORD.

	President Hoover.	*President Coolidge.*
Contract signed	October 26, 1929	October 26, 1929
Keel laid	March 24, 1930	April 21, 1930
Launched	December 9, 1930	February 21, 1931
Sea trials	June 16-20, 1931	September 10-11, 1931
Delivered	July 11, 1931	October 1, 1931
Contract delivery date	October 26, 1931	February 26, 1932

PRINCIPAL HULL DIMENSIONS AND CHARACTERISTICS.

Length overall, feet and inches	654	3
Length on 32-foot water line, feet and inches	630	0
Length between perpendiculars, feet and inches	615	0
Beam, molded, feet and inches	81	0

Depth, molded, to boat deck, at side, feet and inches	79 6
Depth, molded, to promenade deck, at side, feet and inches	70 0
Depth, molded, to bridge deck, at side, feet and inches	61 0
Depth, molded, to shelter deck, at side, feet and inches	52 0
Depth, molded, to upper deck, at side, feet and inches	41 9
Depth, molded, to main deck, at side, feet and inches	32 9
Load draft, normal, molded	32 0
Displacement at load draft, tons	31,063
Tons per inch immersion	94.9
Gross tonnage	21,936
Net tonnage	12,986
Block coefficient at maximum draft	0.675
Mid-section coefficient at maximum draft	0.979
Designed speed in knots	20.5

DESCRIPTION OF HULLS.

The hulls are of the complete superstructure type. There are nine decks in all, five of which are fully plated. Transverse framing is used throughout. The frames amidships are spaced 36 inches and forward and aft are reduced by steps to 24 inches in the peaks. The hulls are subdivided by ten water-tight bulkheads, seven of which extend to the shelter deck, two to the upper deck and the collision bulkhead to the bridge deck. The double bottom extends from the collision bulkhead aft to frame No. 200, and it is divided longitudinally into eleven compartments. Those forward of the engine room are fitted for carrying fuel oil or water ballast and those under and aft of the engine room are fitted for carrying fresh water.

Sixteen water-tight doors are provided in the water-tight bulkheads below the main deck. They are electrically operated and controlled by the Cutler-Hammer–Newport News system.

Each vessel has a slightly raked straight stem and an elliptical stern of the protected rudder type. The stern is bossed to carry the propeller shafting.

There are two masts, each fitted with cargo handling booms, and one smoke pipe. What appears to be a second or after smoke pipe is a dummy for the engine room ventilation.

Accommodations are provided for four classes of passengers as follows:

First class	307
Special class	133
Third class	170
Steerage	378
Total	988

The first class accommodations are located amidships on five decks and served by two Otis electric elevators, each having a capacity of 1800 pounds at a speed of 200 feet per minute. There are 108 staterooms and four de luxe suites. The main dining saloon is on the upper deck adjoining the main entrance lobby. It seats 272 persons and a private dining room adjoining seats 18 more. On the promenade deck are located the writing room, lounge, smoke room, marine tea garden and soda fountain. An open air swimming pool, gymnasium, and children's play room are provided on the boat deck.

There are 39 special-class staterooms on the shelter deck adjoining the first-class quarters. They are arranged to be used as first-class accommodations if so desired. The smoking room and lounge for the special-class are on the bridge deck and the dining saloon, seating 120 persons, is on the upper deck. The cargo hatch forward of the lounge on the bridge deck is fitted for an open air swimming pool for the special-class. A deck promenade is provided abreast the swimming pool, port and starboard.

The third-class passengers are accommodated in 23 staterooms located on the upper deck aft. A dining saloon is also provided on this deck seating 114 persons. Their social hall is located on the shelter deck aft and the after end of the bridge deck provides their open air deck space.

Permanent accommodations for 60 steerage passengers and portable berths for 318 additional are located on the main deck aft. Open air deck space is provided for them on the shelter deck aft.

The officers and crew will number about 323. The deck officers are quartered on the boat deck forward and the engineers on the same deck, starboad side amidship. The petty officers, crew, and stewards have quarters on the shelter and upper decks forward and the engineer's force are provided for on the main deck, amidship, port side.

LIFEBOATS AND LAUNCHING GEAR.

Each vessel carries the following metallic boats:

Number.	*Type.*	*Length.*	*Capacity.*
10	Life boats	32 feet 6 inches	90 persons, each
4	Life boats	28 feet 0 inches	60 persons, each
2	Life boats	26 feet 0 inches	50 persons, each
2	Work boats	26 feet 0 inches	35 persons, each
2	Motor boats	28 feet 0 inches	45 persons, each

The 32 foot 6 inch boats are handled by ten sets of Welin-McLachlan gravity davits, motor or hand controlled. Each motor is 13½ horsepower and of the enclosed water-tight type. The motor boats and other boats are handled by eight sets of hand operated Welin quadrant davits. Two Hyde Windlass Company's electric boat winches are installed for handling the motor boats and work boats. The winch motors are of the enclosed water-tight type of 25 horsepower each.

The above motors, likewise those for the mooring and steering gears, cargo winches, refrigerating plant, and ventilation fans, mentioned in the following paragraphs, were furnished by the General Electric Company for the *President Hoover* and the Westinghouse Electric and Manufacturing Company for the *President Coolidge*.

ANCHOR WINDLASS AND GEAR.

Each vessel has two 18,900 pound bower anchors, one 16,065 pound spare bower anchor, and one 6825 pound stream anchor. Each main bower anchor is provided with 165 fathoms of 3¼-inch stud-link chain cable.

The windlass was furnished by the American Engineering Company. It is located on the forecastle and it is of the horizontal, steam-driven, spur-geared type with 14-inch by 14-inch double-cylinder engine.

MOORING GEAR.

An American Engineering Company's steam capstan is located on the bridge deck forward for mooring. It is of the independent reversible type and has a 26-inch diameter head driven by a 10-inch by 12-inch engine. Three Hyde Windlass Company's electric-

driven reversible capstans are located aft on the shelter deck as part of the mooring equipment. They are each driven by a 75-horsepower enclosed water-tight motor.

STEERING GEAR.

The steering gear is of the American Engineering Company's electro-hydraulic type. It is installed in the steering gear room just forward of the rudder stock. The gear has four cylinders, arranged in parallel pairs fore and aft, and fitted with double-ended rams carrying trunnion blocks at the center, which in turn connect through links to the crosshead on the rudder stock.

The rudder is of the balanced type.

Duplicate pumping sets are provided, each consisting of an electric motor coupled to a Hele-Shaw variable stroke delivery pump. The motors are 75-horsepower.

The steering gear is controlled by a Sperry two-unit gyro-pilot and an hydraulic telemotor of the American Engineering Company's type. They are arranged with an electric clutch for shifting over to the hydraulic telemotor in case of failure of the gyro-pilot gear.

CARGO HANDLING GEAR.

Cargo is carried in six holds, three forward and three aft, having a total capacity of 550,000 cubic feet for general cargo and 59,500 cubic feet for refrigerated cargo. Holds, Nos. 1, 2, 4, 5, and 6, have access through weather deck hatches and hold No. 3 through a combination of side ports and 'tween deck wing hatches. In addition to the access enumerated, hold No. 2 has side ports and 'tween deck wing hatches.

All hatches are served by an elaborate system of booms and winches of the latest type. The weather deck hatches for general cargo holds Nos. 1, 2, 5, and 6 are each provided with four 5-ton cargo booms. Each boom is equipped with a reversible, single-drum, spur-geared winch driven by a 35-horsepower enclosed water-tight motor. In addition, No. 2 hold is served by a 30-ton boom, two of the above mentioned winches being of the single geared, two speed type for use in connection with the latter. The 'tween deck hatches for No. 3 hold are each served by two 4½-ton cargo cranes and four winches driven through double gear

reduction by 25-horsepower motors. These winches also serve four tramway hoists for hoisting cargo from barges alongside through the cargo ports. There are also two Hyde electric automobile handling capstans at the upper deck, hold No. 2, each driven by a 15-horsepower motor. Refrigerated cargo is carried in hold No. 4, which is provided with two wing weather deck hatches, each served by two booms and two winches driven by 35-horsepower enclosed water-tight motors.

All cargo winches were furnished by the Lidgerwood Manufacturing Co.

LAUNDRY EQUIPMENT.

The laundry is located on the main deck, starboard side, abreast the boiler hatch. The following equipment, all furnished by the American Laundry Machinery Company, is installed:

1 Washing Machine, solid-head, two-compartment type, motor-driven.

1 Vertical extractor (drying equipment), motor-driven.

1 Steam-heated, super-suction tumbler (drying equipment), motor-driven.

1 75-inch motor-driven return-apron ironer.

1 Self contained ironing board with electric hand iron equipment.

A storage room is provided for soiled linen and a locker room for clean linen.

GALLEY AND PANTRY EQUIPMENT, ETC.

The galleys for the first-class and special-class passengers are located on the upper deck, amidship in a common compartment. They are equipped with electric ranges of the most modern type together with all the usual auxiliary equipment. The galley for the third-class and steerage passengers is on the shelter deck aft and those for the crew on the upper deck forward in the crew's quarters.

Starboard of the main galleys are the china scullery, butcher's shop, meat box, bread room, and the first class pantries. On the port side are the bakery, tourist's pantry, galley scullery, vegetable room, and glass, china, and silver scullery.

REFRIGERATING PLANT.

The Brunswick-Kroeschell Company furnished the refrigerating plant, which is of the CO_2, brine circulating type. There are four 4-inch × 5¾-inch, vertical, three-cylinder, single-acting, CO_2 compressors, each driven by a 100-horsepower electric motor. Three units are sufficient to maintain the required temperatures in the cold storage spaces with circulating water at not less than 85 degrees F., as obtains in tropical waters, the fourth unit being held as a standby.

Four brine coolers are provided, one for each compressor. They are designed to cool the brine to a temperature of minus 10 degrees F.

The brine is circulated through the cooling coils in the cold storage spaces by three pumps, two of which are of sufficient capacity for full load requirements. A small brine circulating pump is also provided for supplying brine to the ship's cold storage boxes. The latter are also fitted with an emergency brine circulating connection from the main system. Two cooling water circulating pumps are installed for the CO_2 compressors, each capable of carrying the plant at full load. There is also a small pump for supplying ice water for the drinking water system. The pumps are more fully described in the pump table.

There are fifteen refrigerated cargo spaces, twelve of which, about 40,000 cubic feet total capacity, are air-cooled. The cooling air is supplied by motor-driven American Blower Company "Sirocco" fans, one to each compartment, which discharge the air over brine cooling coils in the supply air duct, thence, through the air casing, covering a wall surface of the compartment, and into the room through openings in the sides and bottom of the air casing. The exhaust air is drawn off through openings in a similar air casing on the opposite wall of the compartment and returned to the fans through a duct. The remaining refrigerated cargo spaces and the ship's cold storage rooms, about 20,000 cubic feet in each, are cooled by brine circulating through cooling coils mounted on the wall and overhead.

In addition to the main refrigerating plant, there are three Frigidaire installations, each with a one horsepower compressor.

One is installed in each of the following spaces: First-class beverage room, officers' deck pantry, and the special-class beverage room and pantry.

VENTILATING SYSTEM.

All staterooms and living quarters, including crew's quarters below the boat deck, galleys, pantries, mess rooms, and machinery spaces are mechanically ventilated by means of supply and exhaust systems. The engine room system functions through a ventilating funnel, located over the engine hatch and similar in size and appearance to the smoke pipe.

Motor-driven ventilating fans are provided as follows:

Number.	*Capacity c.m.f. each.*	*Kind.*	*Spaces Ventilated.*
2	6000	Exhaust	Toilets and baths.
4	5000	Supply	First-class state rooms.
2	4000	Supply	First-class state rooms.
1	3200	Exhaust	First-class lounge.
2	4000	Supply	First-class dining room.
1	4000	Exhaust	First-class smoking room.
1	4000	Exhaust	Crew's toilets.
1	6000	Supply	Crew's quarters, mess and laundry.
4	30000	Supply	Engine room.
2	12000	Exhaust	Galleys and pantries.
1	6000	Supply	Galleys and pantries.
1	5000	Exhaust	Toilets.
1	1860	Exhaust	Library.
1	1860	Exhaust	Officers' mess room.
3	6000	Exhaust	Steerage spaces.
3	5000	Supply	Steerage spaces.
2	4000	Exhaust	Steerage spaces.
1	4000	Exhaust	Steerage toilets and hospital.
2	5000	Supply	Crew's quarters.
1	6000	Exhaust	Crew's quarters.
1	6000	Supply	Refrigerating machinery space.
1	4000	Supply	Tourists' dining room.
1	3000	Supply	Tourists' state rooms and hospital.
39			

The B. F. Sturtevant Co. furnished all the ventilation fans.

HEATING SYSTEM.

All living quarters and spaces, except the first-class dining saloon, are heated by steam radiators. The latter is heated by steam coils in the ventilation ducts. All first-class and special-class staterooms and bathrooms are heated by electric heaters.

FIRE PROTECTION.

The Walter Kidde and Company installed the fire protection equipment consisting of the " Selex" thermostatic fire alarm system with alarm bell signal and indicator in the wheel house for all passenger and crews quarters, baggage rooms, mail room, and adjacent stores and linen rooms, the " Rich" system for fire detection for all cargo spaces, paint and lamp rooms, with steam smothering pipes led to the bottom of all cargo spaces, etc., and the " Lux" CO_2 fire-extinguishing system for the boiler rooms with distributing pipes below the floor plates and hose connections above the floors.

FIRE MAIN.

In addition to the fire protection system described above, the usual salt water fire main extends throughout the vessel with hose valves at convenient locations on the various decks for fighting fires. The main is supplied by either of four pumps located in the engine and fire rooms as listed in the pump table.

SANITARY SYSTEM.

The sanitary system comprises separate fresh-water and salt-water systems connected to all pantries, toilet rooms, etc., as required for comfort and convenience. The cold fresh-water is supplied by a gravity tank which is kept filled by reciprocating pumps. Hot fresh-water is taken from the cold water supply and heated by a steam coil in a pressure tank in the engine room. The temperature of the water is controlled by temperature regulators. From the pressure tank mains are led to the various decks with branches as required. An ice water system is also installed with outlets at convenient locations for service and the comfort of passengers.

The cold salt water system is supplied by two motor-driven centrifugal pumps and a reciprocating pump as auxiliary source of supply. Mains are led to the various decks with branches as required. Hot salt water is supplied from a pressure tank in the engine room similarly fitted with steam heating coils and temperature control as for the fresh water system.

The fresh water is carried in deep tanks and inner bottom, having a total capacity of 1908 tons, exclusive of reserve feed.

DRAINAGE SYSTEM.

A drainage main extends from the forward pipe-tunnel through the boiler, engine, and refrigerating machinery rooms to the shaft alleys. It has suction connections to the various pumps noted in the pump table. The bilge water is pumped overboard direct or through a 75-ton Bethlehem oil and water separator on the main deck. which removes the oil from the water and returns it to a tank while the oil cleansed water is discharged overboard.

Two of the main circulating pumps in the engine room are provided with bilge suctions, fitted with nonreturn stop valves. These connections are for use in extreme emergencies only.

TRIMMING TANKS.

The forward and after peak tanks, inner bottom tanks Nos. 1 to 7, inclusive, and the fuel oil deep tanks are arranged for flooding and pumping as required to maintain the vessels in proper trim.

RADIO EQUIPMENT.

Wireless apparatus is provided of sufficient range to keep the vessels in touch with the company's main office in San Francisco from any part of the world.

The transmitting apparatus is provided in duplicate.

A 140-ampere-hour, 240-volt, 120-cell Exide storage battery is provided as final reserve for the radio.

In addition to the above equipment, forty loud speakers are distributed at convenient locations in all public spaces of the first, special, and third classes for transmitting radio broadcast reception programs or phonographic records. The apparatus for this service is of the Radio Corporation of America's make on the *President Hoover* and the Western Electric Co. on the *President Coolidge*.

NAVIGATING EQUIPMENT.

There are three steering stations, the regular station in the wheel house and emergency stations on top of the wheel house and on the bridge deck aft. Standard magnetic compasses are provided on top of the wheel house and in the after steering station, while a magnetic spirit compass is installed in the wheel house. All compasses and binnacles are Navy Standard type furnished by John E. Hand and Sons Company.

In addition to the above, a Sperry gyroscopic compass system is furnished.

A double-unit Sperry gyro-pilot is also installed in the wheel house for automatic steering. There is also a Sperry course recorder in the chart room, a Kolster radio compass or direction finder, and a Submarine Signal Company fathometer. A Louis Weule Company electric sounding machine is installed on the boat deck, just abaft the navigating bridge.

Two searchlights are provided, one a Sperry 18-inch incandescent type and one a Sperry special 24-inch unit complying with the Suez Canal regulations.

MAIN PROPELLING MACHINERY AND AUXILIARIES.

Both vessels are twin screw, electric drive. Each installation consists essentially of two turbo-driven alternators and two synchronous-induction motors, together with their condensers and the usual auxiliaries. The main turbo-generator sets, main propelling motors, and, in general, all motors driving auxiliary units were manufactured by the General Electric Company and the Westinghouse Electric and Manufacturing Company, respectively, for the *President Hoover* and the *President Coolidge*. In other particulars the machinery plants are alike for both vessels, except as noted in the following brief description.

MAIN TURBO-GENERATOR SETS.

The *President Hoover* has two turbines of the General Electric, Curtis, horizontal type, each direct connected to an alternating-current generator. Each turbine has 16-stages and is designed for 13,250 shaft horsepower at 2660 revolutions per minute with steam pressure at the throttle valve of 275 pounds per square inch

gage and 200 degrees superheat, and a vacuum at the condenser of 28.5 inch. The generators are two-pole, 3-phase alternating-current, each rated 10,100 kilowatts, 4800 volts at 2660 revolutions per minute.

The turbine rotor shaft and all stage wheels, except the first, are one solid forging milled out to form the shaft and wheels, which provides a more rigid and superior construction to the earlier designs with separate wheels keyed to the shaft. All wheels have a single row of blades except the first stage, which has two rows. The blading is rugged and designed to withstand the most severe working stresses with ample margin of safety. The critical speed of the turbine is well in excess of the maximum running speed. The turbine casings are of cast steel, divided longitudinally with hinged upper halves to permit ready inspection of the rotors and interior of the casings. The turbines are mounted directly above their condensers on the upper engine room level. The turbines operating in one direction only, reversal of the propellers being accomplished through the electrical hook-up.

Steam is admitted to the turbine through a strainer and throttle valve, thence to the steam chest in which are located the control valves, which are under the control of the governor and emergency trip gear, and hydraulically actuated, through a mechanical mechanism, by pressure supplied from the lubricating oil system.

The main generators are of the self ventilated enclosed type, the air being passed through separate coolers, under the base of each unit, and recirculated through the windings, thus assuring clean cool air for ventilating purposes. The air is circulated by fans mounted on the end of the rotor shaft and the circulating water is supplied from the main circulating pumps. Ample safeguards are provided against the admission of water into the windings, through tube leakage in the coolers. The rotor is solid forged steel in which slots are milled for the field windings. The stator frame is of steel plate construction and the core is built up of laminated plates fitted with slots for windings. Electric heaters are provided inside the end shields to guard against damage to the windings by moisture when the generator is shut down. Fire extinguishers and provisions for ascertaining the temperature of the armature windings and the field are also provided.

Excitation for the main generators is supplied from any one of the four 500-kilowatt, direct-current generators. For this purpose selective switches are provided and so interlocked that only one of the four units can be used for excitation at any time.

The *President Coolidge* is equipped with two Westinghouse Electric and Manufacturing Company's horizontal combination impulse and reaction turbines, each having one impulse stage of two rows of moving blades and one row of stationary blades at the high pressure end followed by 29-rows of single flow reaction blading exhausting into the condenser. Each turbine is direct connected to an alternating-current generator of the 2-pole, 3-phase type, rated at 10,200 kilowatts, 4000 volts at 2660 revolutions per minute.

The turbines are designed for 13,250 brake horsepower at 2660 revolutions per minute, with 275 pounds gage steam and 200 degrees F. superheat at the inlet nozzles and 28½ inches of vacuum at the condenser. Steam enters the turbine through a strainer and oil-operated throttle valve. Each turbine has an oil-operated governor and an automatic stop governor. The former functions on the variation in oil pressure produced by changes in speed of an impeller mounted on the turbine rotor. The speed of the propelling motor is controlled by varying the speed of the turbine, which is accomplished by an adjustable-speed mechanism connected with the turbine governor and consisting of a relief valve located on the main control board and operated by a lever. A small pipe conveys the oil from the governor to the relief valve, the latter discharging through a drain pipe to the sump tank.

The main generators are of the self-ventilated enclosed type with air coolers mounted beneath them, the system functioning in a similar manner to that outlined for the *President Hoover*. They are designed so that their first critical speed will be above the maximum operating speed.

Excitation is provided from any one of the four 500-kilowatt, direct-current generators as in the case of the *President Hoover*.

MAIN PROPELLING MOTORS.

The *President Hoover* has two main propelling motors, one direct-connected to each line of shafting. They are of the 3-phase, 44-cycle, 40-pole, 4800 volts, alternating-current, synchronous-

induction type, each having a maximum continuous rating of 13,250 shaft horsepower at 133 revolutions per minute.

The stator frames are built up steel plate construction and the rotor spiders are cast steel. The induction motor windings are of the squirrel-cage type and are imbedded in the pole faces of the field cores. They are short circuited by means of end rings and are used only for starting.

Stator temperature detectors are provided, also heating coils for preventing condensation on the windings when shut down.

Each motor is cooled by a separate ventilating fan driven by a direct-current motor rated 22/40 horsepower, 600/740 revolutions per minute at 230 volts.

In the case of the *President Coolidge,* each main propelling motor is of the 3-phase, 44-cycle, 40-pole, 4000-volt, synchronous-induction type, rated 13,250-shaft horsepower at 133 revolutions per minute. The motors are self-contained units, the rotors being carried in sleeve-type bearings supported by heavy end brackets secured to the motor frames. Each motor is separately ventilated by motor-driven propeller-type blower mounted in the air duct above the motor.

CONTROL EQUIPMENT AND OPERATION.

The control platform is located on the upper engine room level, abaft the main turbo-generators.

The control panel contains a complete set of instruments, gages, etc., connected to the main propelling machinery and principal auxiliary units, which enables the engineer in charge to visualize at a glance the operation of the entire plant from the control platform. There are also mounted on the control panel the various levers for maneuvering the ship.

Normally each main turbo-generator furnishes power for driving the main propelling motor on its side of the ship but provision is made for driving at reduced speed both main propelling motors by either main turbo-generator. Cross tie busses with switches are provided to permit the latter.

Variations in propeller speed in either direction is obtained by varying the turbine speed, and is directly under the control of the operator at the control panel. The speed of either propeller may

be varied or reversed irrespective of the other except when one turbo-generator is used for driving both propelling motors, in which case both motors operate at the same speed, though rotating in the same or opposite directions.

The governor control mechanism adjusts the speed of the main turbo-generators to obtain the desired propeller speed, which is one-twentieth of the turbine speed.

Excitation is applied to the generators and motors by means of the field levers.

All the control levers are interlocked to guard against improper operation.

PROPELLER SHAFTING.

Each line of shafting consists of one propeller shaft, supported by two lignum vitae bearings; six sections of line shafting, each supported by two spring bearings; and a thrust shaft, supported by the thrust bearing. All sections of shafting are bolted together by flange couplings, forged solid with the shafts, except the inboard coupling on the propeller shaft, which consists of the usual sleeve-type, keyed to the shaft and through bolted to the line shaft coupling.

The shafting is solid and of the following principal dimensions:

Propeller shaft, diameter, inches	20½
inboard coupling sleeve, diameter, inches	35½
length, inches	16
inboard coupling half-rings, thickness, inches	4
Line shaft, diameter, inches	19
Thrust shaft, diameter, inches	21¼
thrust collar, diameter, inches	47½
Flange couplings, diameter, inches	35½
thickness, inches	4¾
Coupling bolts, number per coupling	8
diameter, inches	3⅞

THRUST BEARINGS.

One thrust bearing is provided for each line of shafting. They are of the usual Kingsbury, pivoted, segmental block type.

TURNING GEAR.

Each line of shafting is provided with a turning gear, consisting of a worm wheel mounted on the flange coupling of the main motor shaft and meshing with a worm driven by a ten horsepower motor.

PROPELLERS.

There are two three-bladed propellers, of the built up type, outboard turning when going ahead. The blades are of manganese bronze and the hubs of semi-steel.

The principal characteristics are given below:

Diameter, feet and inches	18	6
Pitch, feet and inches	18	0
P.A. / D.A.	0.35	
R.P.M., at 20.5 knots, designed	133	

MAIN CONDENSING APPARATUS.

There are two main condensers, one for each main turbine. Each condenser is directly under its turbine, permitting free drainage of all condensate from the turbine to the condenser. The condensers have 14,000 square feet of cooling surface each, are of the two pass type installed with tubes athwartship.

Circulating water is supplied each condenser by duplicate motor driven centrifugal pumps of capacities as given in the pump table. Each pump is capable of supplying sufficient circulating water to its condenser for full power conditions. The pumps are provided with high and lower injection sea chests, the former for use in harbor and the latter at sea.

The condensate is drawn off by three motor-driven condensate pumps, one for each condenser, the third a standby connected to both condensers. The pump particulars are given in the pump table.

Westinghouse air ejectors are provided in duplicate for each condenser. They are of the two-stage type mounted on a combined inter-and-after condenser. Each ejector set has sufficient capacity to remove the air from its condenser under full power operation of the main machinery plant.

BOILER FEED SYSTEM.

The boiler feed system in the engine room consists essentially of a feed and filter tank of 5000 gallons capacity, two main feed pumps and one auxiliary feed pump, a grease extractor, and two Davis Engineering Corporation feed water heaters operating on the two stage principle, and having a combined capacity of 305,000 pounds of water per hour from 110 to 304 degrees F. There are also two additional auxiliary feed pumps, one in each boiler room. The pumps' characteristics are given in the pump table.

The feed water heaters are arranged in series. The first stage heater is supplied with steam at 10 to 12 pounds per square inch gage as heating agent, bled from the lower stages of the main turbines, from the auxiliary exhaust main and from the evaporators, and raising the temperature of the water to 230 degrees F. The second stage heater takes its steam from the intermediate stages of the main turbines at 75 to 80 pounds per square inch gage, raising the water to the final temperature of 304 degrees F. Each heater is fitted with an automatic drain trap discharging to the feed and filter tank. The steam supply lines from the main turbines to the heaters are fitted with automatic check valves, which close when the turbines are stopped by hand or tripped by the governor, thus preventing the admission of steam to the turbines through the connections when the turbines slow down or stop. The check valves are actuated by the governor oil pressure, being held open when the pressure is on and closed when it decreases or fails.

The boilers are normally fed by the main feed pumps, each of which have a separate suction from the feed and filter tank and discharge in sequence through the grease extractor, first and second stage feed water heaters and main feed line to all boilers. The grease extractor and feed water heaters are fitted with by-passes. The auxiliary feed pump in the engine room also has a discharge connection to the main feed system similar to the main feed pump.

The auxiliary feed line is primarily supplied by the three auxiliary feed pumps, which take their suctions from the feed and filter tank. It is arranged to feed all boilers direct or through the feed water heaters. The main feed pumps have discharge connections to the auxiliary feed main.

The main and auxiliary feed lines are cross-connected.

Make up feed is supplied by a connection to each condenser from the reserve feed suction main and also by the auxiliary feed pumps which have suction connections to all reserve feed bottoms and discharge to the boilers.

The reserve feed bottoms have a total capacity of 411 tons of water.

LUBRICATING SYSTEM.

The lubrication of the bearings for the main propelling units is supplied by two lubricating oil pumps as listed in the pump table. The pumps deliver the oil through coolers to a gravity tank in the engine hatch, which supplies all bearings, draining to the sump tanks from which the pumps take their suction.

The required lubricating oil pressure for operating the turbine governors, etc., is obtained by a back pressure valve fitted in the pump discharge line and set at the desired pressure. The governor oil line is taken off the discharge main ahead of this valve.

One lubricating oil purifier is provided for cleansing the oil.

WATER SERVICE.

A water service system is provided for the main thrust bearings, all shaft steady bearings, and for flushing the stern tube bearings. It is supplied from the salt water sanitary system.

BOILERS.

There are twelve oil burning boilers of the Babcock and Wilcox water-tube, header type, each fitted with a superheater in the first pass. They are installed in two compartments of six boilers each and arranged in two athwartship batteries of three boilers each, facing each other with a common operating flat between and extending the width of the vessel.

Desuperheating coils are fitted in the drums of the three forward and the three after boilers for supplying steam to the saturated steam auxiliaries.

The boilers are designed to generate the necessary steam for the full power requirements of the machinery plant, with an average air pressure of about one inch of water at the fuel oil burners.

The boiler characteristics are given below:

Working pressure, pounds per square inch	300
H.S.. generating tubes, total, square feet	57,624
superheaters, total, square feet	5,700
Superheat, degrees F	200
Number of oil burners, each boiler	4
Type of oil burners	Cuyama

UPTAKES AND SMOKE PIPES.

The uptakes are of the usual design, all leading into a common elliptical smoke pipe, major axis 21-feet and minor axis 14-feet, by 121 feet 1¾ inches high above the center of the oil burners. The smoke pipe is subdivided by division plates extending to the top into three cross sectional areas. The after smoke pipe is a dummy, serving for ventilation only.

FORCED-DRAFT BLOWERS.

Two motor-driven, horizontal forced draft blowers are provided for each boiler room. They are located at the working level, outboard of the boilers, one port and one starboard. They take suction from the boiler room and each discharge through a separate athwartship duct, led under the floor plates, to a battery of boilers, the two ducts in a fire room being cross-connected so that either blower may supply all boilers, or combinations thereof, in emergencies or at low powers.

Air is supplied each fire room through two large ventilator trunks terminating at the sun deck in large cowls arranged for training into the wind.

The blowers are of the following characteristics:

Type	Sturtevant, No. 90, silent-vane, horizontal
Capacity, C.F.M., each	30,000
Air pressure, inches of water	2
R.P.M.	850
Horsepower of motors, each	20

FUEL OIL SYSTEM.

The fuel oil system consists of two transfer pumps, two main service pumps, two auxiliary service pumps, and a pump for

removing sediment from the fuel oil tanks, all as listed in the pump table, also six fuel oil heaters, furnished by the Griscom-Russell Company; four of which are capable of handling the fuel oil at full power conditions. The usual appurtenances such as strainers, meters, etc., are fitted.

The service pumps, two required for full power, take suction from the deep tanks and discharge to the oil burners through strainers and heaters. They are also arranged to take suction from the fuel oil transfer suction main.

The transfer pumps are arranged to pump oil from any of the inner bottom and deep storage tanks to any other deep storage tank, direct or inner bottom via deep tank. They also have a discharge connection to the filling line for transferring oil to another vessel or to the dock.

The fuel oil storage tanks have a total capacity of 610 tons.

MAIN STEAM PIPING.

There are four 8-inch main steam lines, two port and two starboard, each supplied by a battery of three boilers through 5-inch branch pipes from the superheaters. The two port lines join and form a common 10-inch line in the engine room, which is led to the port main turbine throttle valve, and the two starboard lines are similarly connected to the starboard main turbine throttle valve. The two 10-inch lines in the engine room are cross-connected.

AUXILIARY STEAM PIPING.

An auxiliary steam line for superheated steam is fitted in the engine room for supplying steam to the auxiliaries using superheated steam. It is supplied from all four of the 8-inch main steam lines, in order that superheated steam may be available for this service under all conditions of operation.

A saturated steam line is also provided for the auxiliaries operating on saturated steam. It extends the full length of the engine and boiler rooms and is supplied from the desuperheaters fitted in the drums of the three forward boilers in the forward room and the three after boilers in the after room.

AUXILIARY EXHAUST PIPING.

An auxiliary exhaust line, connecting to all auxiliaries, extends throughout the machinery spaces. It operates at about ten pounds per square inch gage back pressure, and is connected to each of the two auxiliary condensers through spring loaded back pressure valves and the first stage feed water heater through a stop valve. It is connected also to the atmosphere.

ESCAPE PIPES.

All the boiler safety valves and superheater safety valves have their outlets piped to an escape pipe led to the atmosphere on the after side of the smoke pipe.

INTERIOR COMMUNICATION.

The customary engine and fire room telegraphs, gongs, telephones, voice tubes, etc., are fitted for transmitting orders and signalling from the bridge to the engine room, and between the various machinery compartments.

AIR COMPRESSOR PLANT.

An Ingersoll-Rand, horizontal, single-cylinder, single-stage air compressor is installed for furnishing compressed air for pneumatic tools, cleaning boilers, generators and motors and the pneumatic tube system, which transmits messages from the radio room to the purser.

The compressor is belt driven by an 85 horsepower motor and its cylinder is 14-inches diameter by 12-inches stroke. It supplies air at about 100 pounds per square inch gage.

One accummulator tank is provided.

EVAPORATING AND DISTILLING PLANT.

The evaporating and distilling plant is single effect, operating on low pressure steam bled from the intermediate stages of the main turbines at about 70 pounds pressure gage. A live steam connection is also provided for emergency use. The plant consists of three Griscom-Russell Company Reilly evaporators, each of 50-tons capacity of distilled water per 24-hours, and two Reilly

distillers, each of 10,000 gallons capacity of fresh water per 24-hours, and one horizontal evaporator feed pump as listed in the pump table. The evaporators and the feed pump are located in the lower engine room, starboard side, and the distillers are in the engine room hatch, at the main deck level, starboard side.

ENGINEERS' WORKSHOP.

A well equipped engineers' workshop is located at the lower deck level, starboard side of the engine room, in a space separated from the engine room by a screen bulkhead.

It has the following equipment:

1 Cincinnati emery grinder, 2-wheels, 2 inch face by 12 inch diameter, 2-horsepower motor.

1 Braves drill press, 1-horsepower motor.

1 U. S. Machine Tool Co. sensitive drill, ½-horsepower motor.

1 South Bend tool room lathe, 15-inch swing by 5-foot bed, 1-horsepower motor.

1 Rahn-Larmon extension gap lathe, 3-horsepower motor.

1 Western Machine Tool Works 16-inch step toe shaper, 3-horsepower motor.

1 Work bench with two 6-inch machinist's vises.

ELECTRIC LIGHTING AND AUXILIARY POWER PLANT.

The ship's lighting and auxiliary power plant consists of four modified 3-wire type, direct current, 500 Kw., 120/240 volts, turbo reduction gear generator sets, located in the main engine room, upper level, port side. The turbines are designed to operate at a steam pressure of 300 pounds per square inch gage and 200 degrees F. superheat, exhausting into a vacuum of about 28-inches. One set is also arranged with an atmospheric exhaust connection for use in dry dock or in emergencies. Any one of the sets may be used for excitation of the main alternators, as stated elsewhere, while the others are used for the ship's service. The General Electric Co. supplied the units for the *President Hoover*, while those for the *President Coolidge* are of Westinghouse design.

The generator sets are arranged in pairs, the turbines for each pair exhausting downward into a condenser below. Each condenser has 2800 square feet of cooling surface and circulating

water is supplied by a centrifugal motor driven pump. Three motor driven centrifugal condensate pumps are provided, one for each condenser, the third a stand-by for use on either condenser. The pumps are described in the pump table. Twin air ejectors with inter-and-after condensers are provided for each condenser, similar to those for the main condensers.

The direct-current switchboard is abaft the generators, at same level and extends athwartship. It is about 30 feet long and 7 feet high and consists of thirteen panels. It comprises two independent bus systems, one for excitation of the main propulsion generators and propulsion auxiliaries and the other for general power and lighting. Certain switches are double throw to feed from either bus system.

Twelve power distribution panels are installed in various parts of the ships for the control of auxiliaries.

In addition to the above enumerated equipment, two 15-kilowatt, 120/240-volts, 3-wire, direct-current generators, each driven by a Buda gasoline engine, are installed on the boat deck amidship, as emergency source of supply.

TRIALS.

The trials of the *President Hoover* were conducted last June. They consist of a standardization trial over the U. S. Navy measured mile, off Rockland, Maine, and an eight hour economy trial at sea at rated shaft horsepower of 26,500.

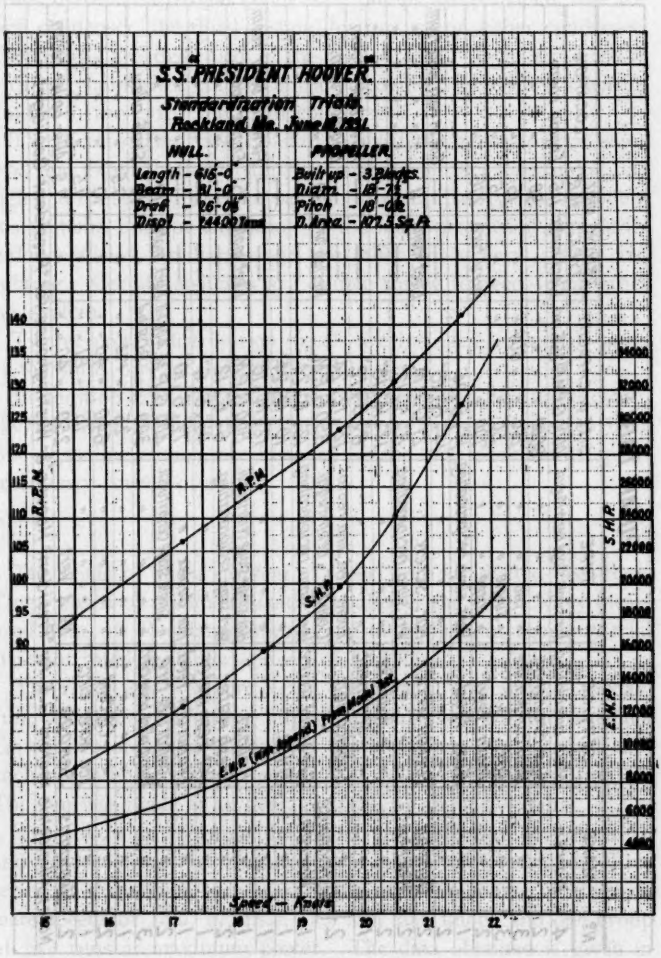
The vessel left Newport News, Virginia, at 5.00 A.M., Tuesday, June 16, 1931, and after adjusting compasses off the Virginia Capes, proceeded to Rockland, Maine, where she arrived and anchored off the breakwater at 6.45 P.M., Wednesday, June 17. The run was made at an average speed of about 20-knots.

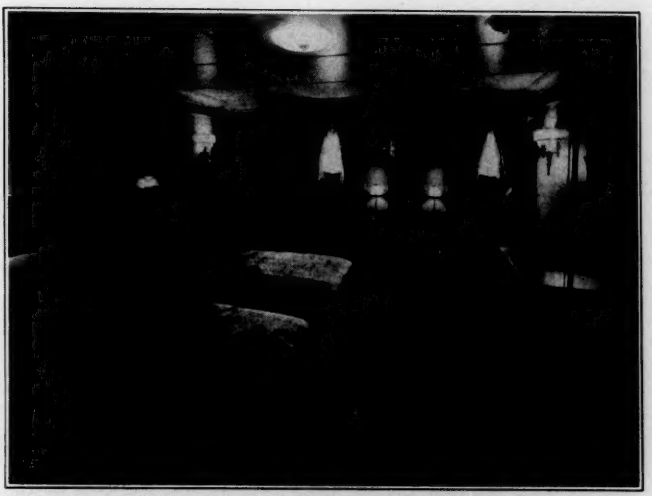
The following morning the vessel got underway at 4.00 A.M. and proceeded to the measured mile course off Rockland. Five series of three runs each were made over the course at approximate speeds of 15.5, 17, 18.5, 19.5 and 20.5 knots, and one group of five runs at maximum power. The data obtained is given in the tabulation, from which the curves were plotted. The trial was concluded at 2.20 P.M. and the vessel headed to sea for the eight-hour economy run.

PUMP TABLE.

Nº	MAKE.	SERVICE	TYPE	SIZE or CAPACITY.	DRIVING TYPE.	UNIT H.P.	LOCATION.
4	Warren.	Main circulating.	Hor. single stage centrifugal.	10000 G.P.M. @ 25ft. hd.	Elect. motor.	115	Engine room
2	"	Aux. circulating.	" " " "	3600 " " " "	"	30	" "
3	"	Main condensate.	" two " "	250 " " 60ft. "	"	15	" "
3	"	Aux. "	" Single " "	50 " " " "	"	5	" "
2	"	Main feed.	" three " "	700 "	Terry turb.	276	" "
1	"	Aux. feed.	Vert. simp. reciprocating.	14" x 8" x 24"	——	——	" "
2	"	" "	" " "	14" x 8" x 18"	——	——	One in each boiler room.
2	Quimby.	Fuel oil service.	Hor. Screw.	27 G.P.M.	Elect. motor.	11	" " " " "
2	Warren.	" " "	Vert. simp. reciprocating.	8" x 5" x 12"	——	——	" " " " "
2	"	" " transfer.	" " "	12" x 14" x 18"	——	——	" " " " "
1	"	" " sediment.	" " "	6" x 6" x 12"	——	——	After boiler room.
2	"	Lubrication.	"Hoover" vert. simp. recip.	12" x 12" x 24"	——	——	Engine room.
2	Westinghouse.	"	"Coolidge" " centrifugal.	350 G.P.M.	Westgh. turb.	30	" "
1	Warren.	Fire & gen. service.	Vert. simp. reciprocating.	12" x 8" x 24"	——	——	" "
1	"	Bilge.	" " "	10" x 12" x 24"	——	——	" "
1	"	"	" " "	12" x 14" x 18"	——	——	Forward boiler room.
1	"	Ballast.	Hor. single stage centrifugal.	800 G.P.M.	Elect. motor.	25	Engine room.
2	"	Fresh water.	Vert. simp. reciprocating.	6" x 6" x 12"	——	——	" "
1	"	Hot fresh water.	" " "	4½" x 5" x 6"	——	——	" "
1	"	Evaporator feed.	Hor. duplex "	7½" x 6" x 10"	——	——	" "
2	"	Sanitary & fire.	" single stage centrifugal.	400 G.P.M. 40 ft. hd.	Elect. motor.	10	" "
3	"	Refrig. plant brine circ.	" " " "	300 " 80ft. hd.	"	16	Shaft alley.
1	"	" " " "	" " " "	25 " 179 " "	"	7.5	" "
2	"	" "CO₂ Cond."	" " " "	640 " 40 " "	"	10	" "
1	"	Ice water circ.	" " " "	15 " 81 " "	"	1.5	" "
2	Yeomans.	Sewage	Nº 4 Vert.	250 " 40 " "	"	7.5	After boiler room.

Note: All electric motors G.E. for "President Hoover" & Westinghouse for "President Coolidge."


S.S. PRESIDENT HOOVER.
Standardization Trials.
Rockland, Me. June 12, 1931.
HULL.
Length - 616-0"
Beam - 81-0"
Draft - 26-0"
Displ - 24400 Tons
PROPELLER.
Built up - 3 Blades
Diam. - 18-7½
Pitch - 18-0½
D. Area - 177.5 Sq. Ft.
R.P.M.
S.H.P.
E.H.P.
R.P.M.
S.H.P.
E.H.P. (Ref. Hovgard) from Model Test
Speed — Knots



SPECIAL SUITE STATEROOM—"PRESIDENT HOOVER."

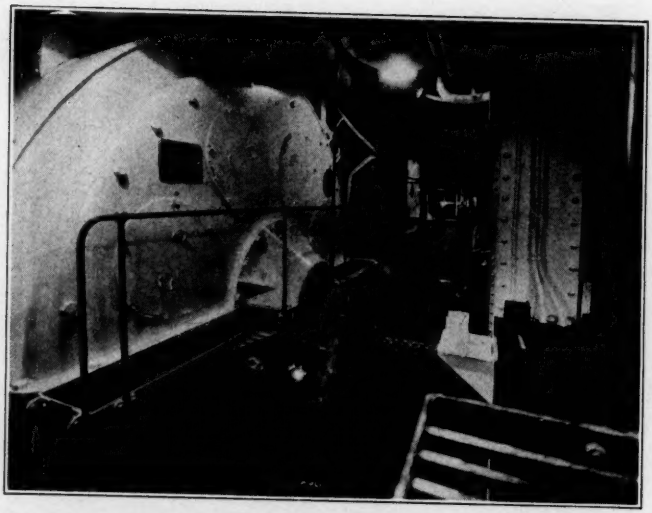
SPECIAL SUITE SITTING ROOM SHOWING SUN PORCH—"PRESIDENT HOOVER."

1ST CLASS DINING ROOM—"PRESIDENT COOLIDGE."
(ANOTHER VIEW.)

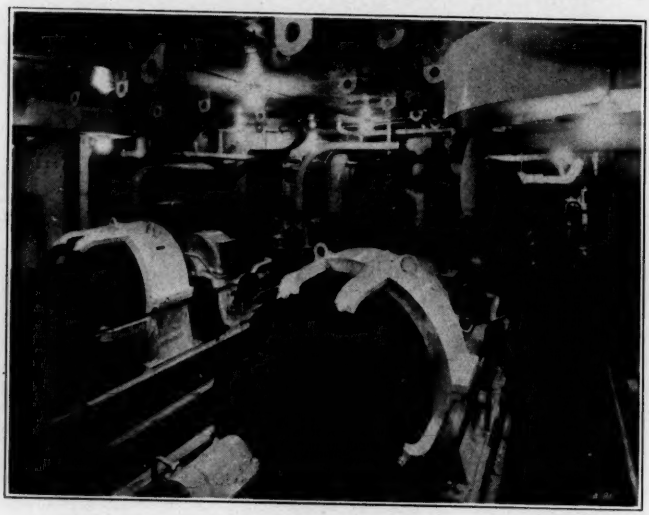
FIRST CLASS LOUNGE—"PRESIDENT HOOVER."

First Class Smoking Room—"President Hoover."

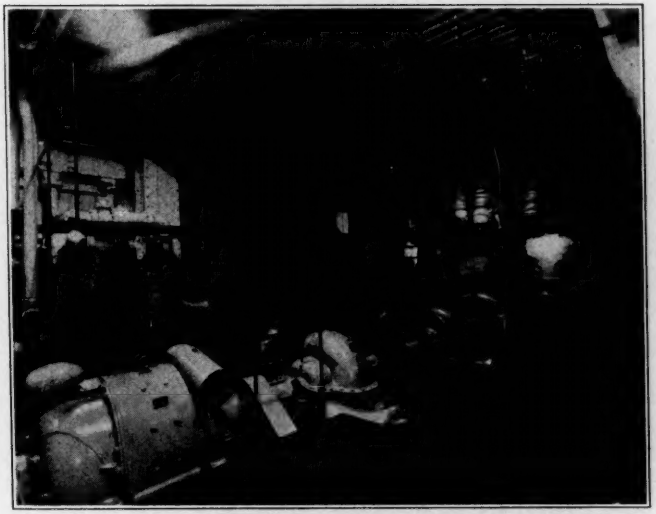
Upper Engine Room—"President Hoover."



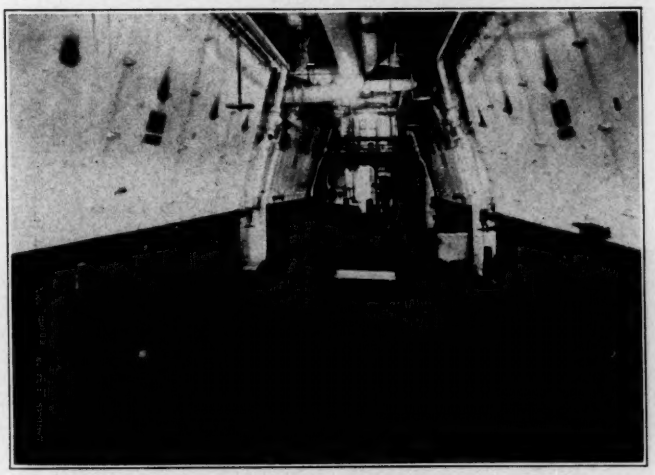
LOWER ENGINE ROOM—"PRESIDENT HOOVER," SHOWING MAIN MOTORS.



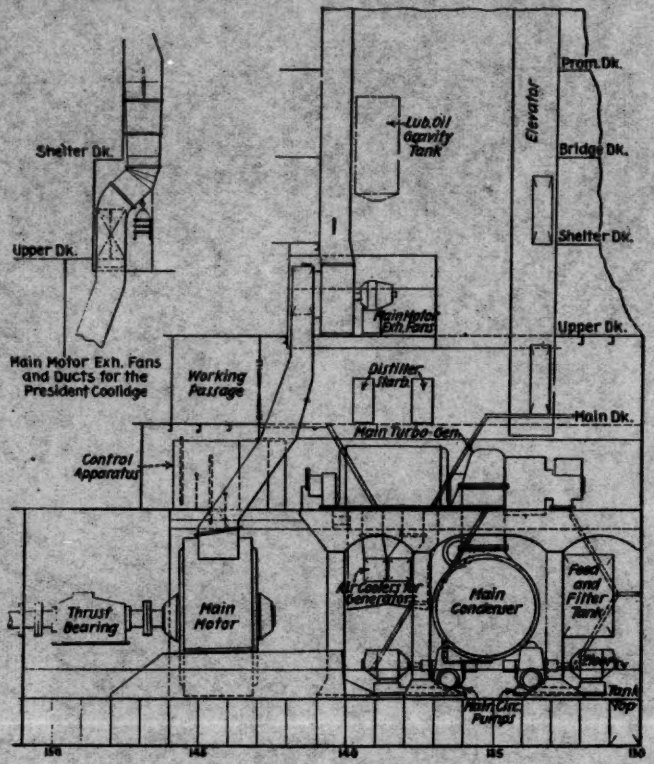
AUXILIARY GENERATORS—"PRESIDENT COOLIDGE."



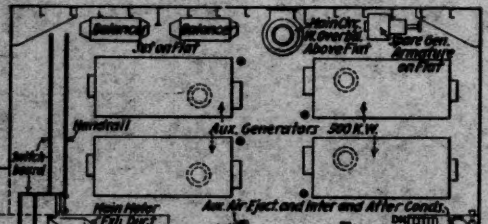
FORWARD END—LOWER ENGINE ROOM—"PRESIDENT HOOVER."

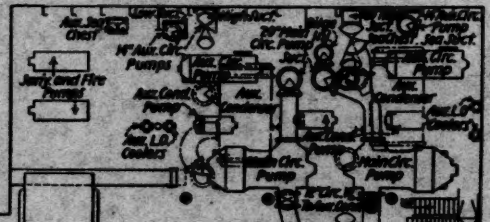


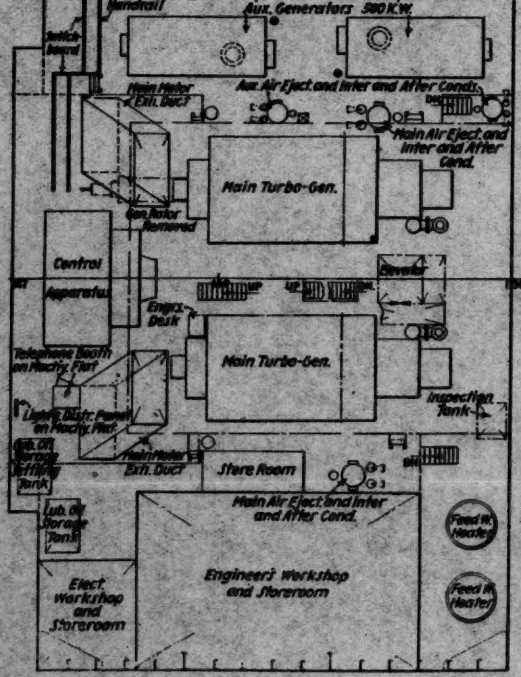
BOILER ROOM—"PRESIDENT HOOVER."



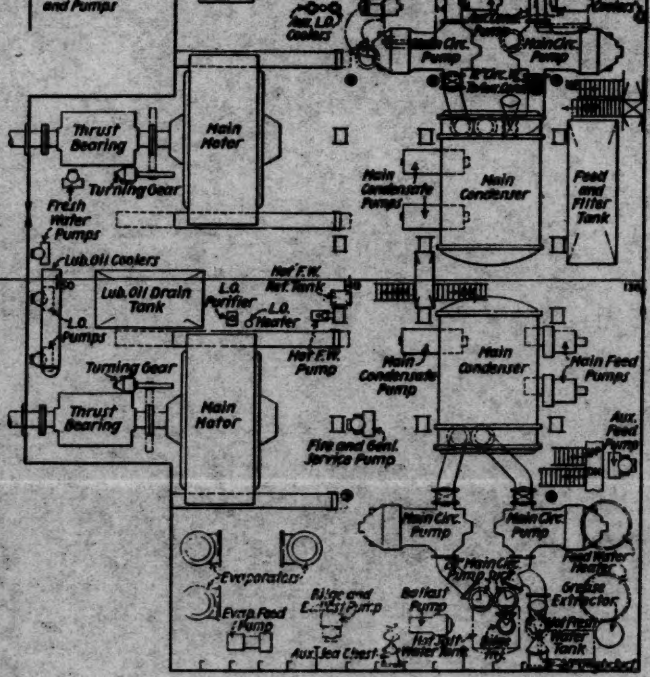
Elevation of engine room



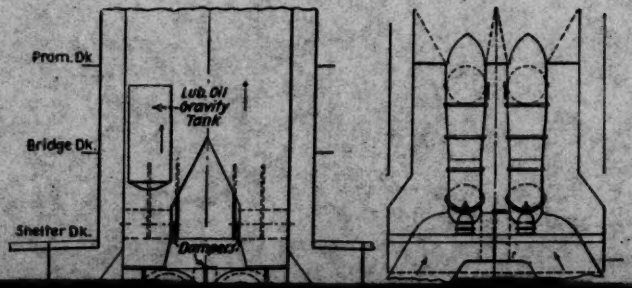


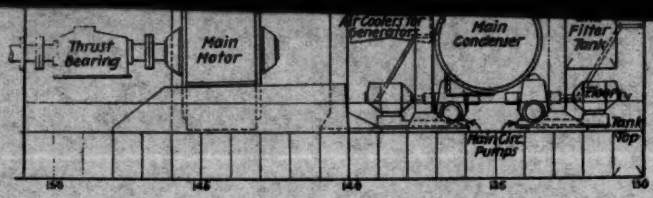


Plan of machinery flat

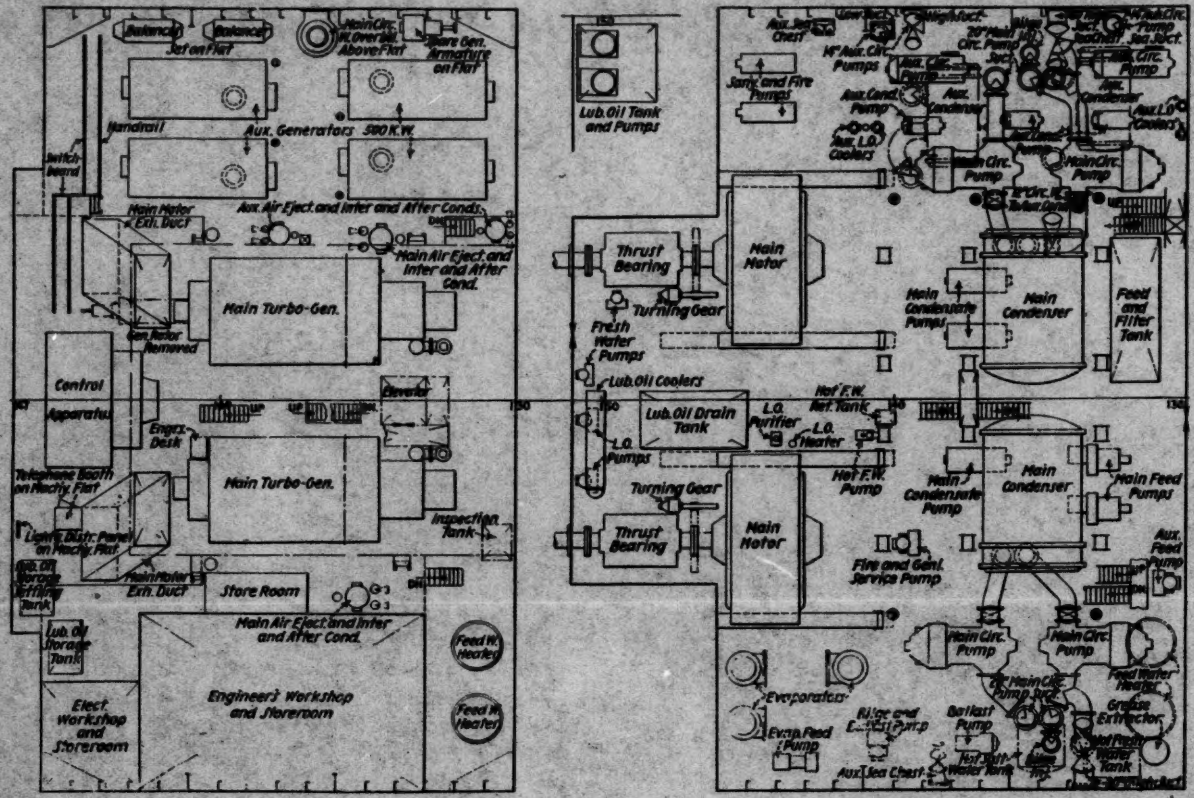


Plan of engine room





Elevation of engine room

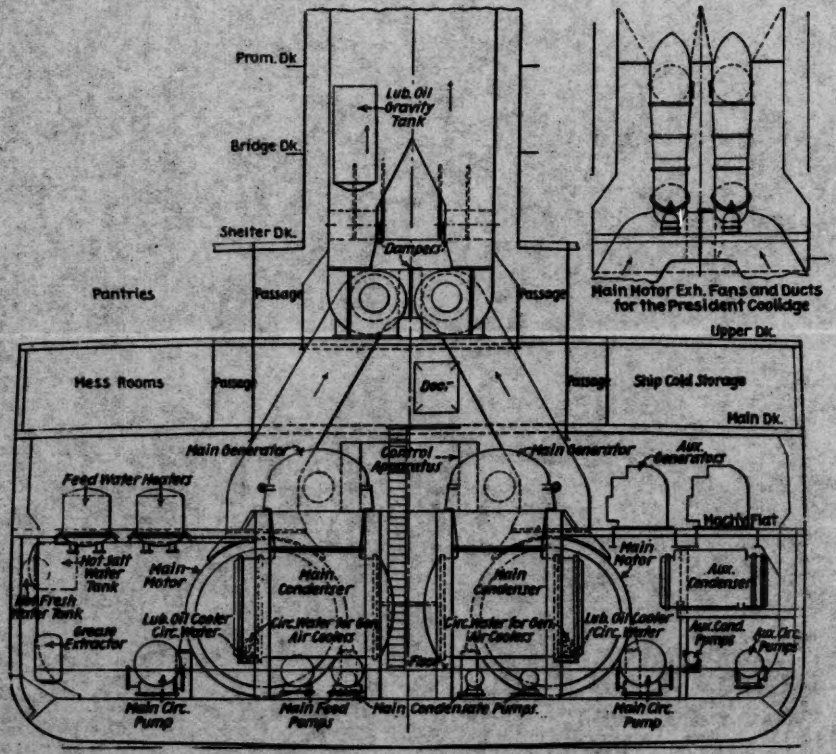


Plan of machinery flat

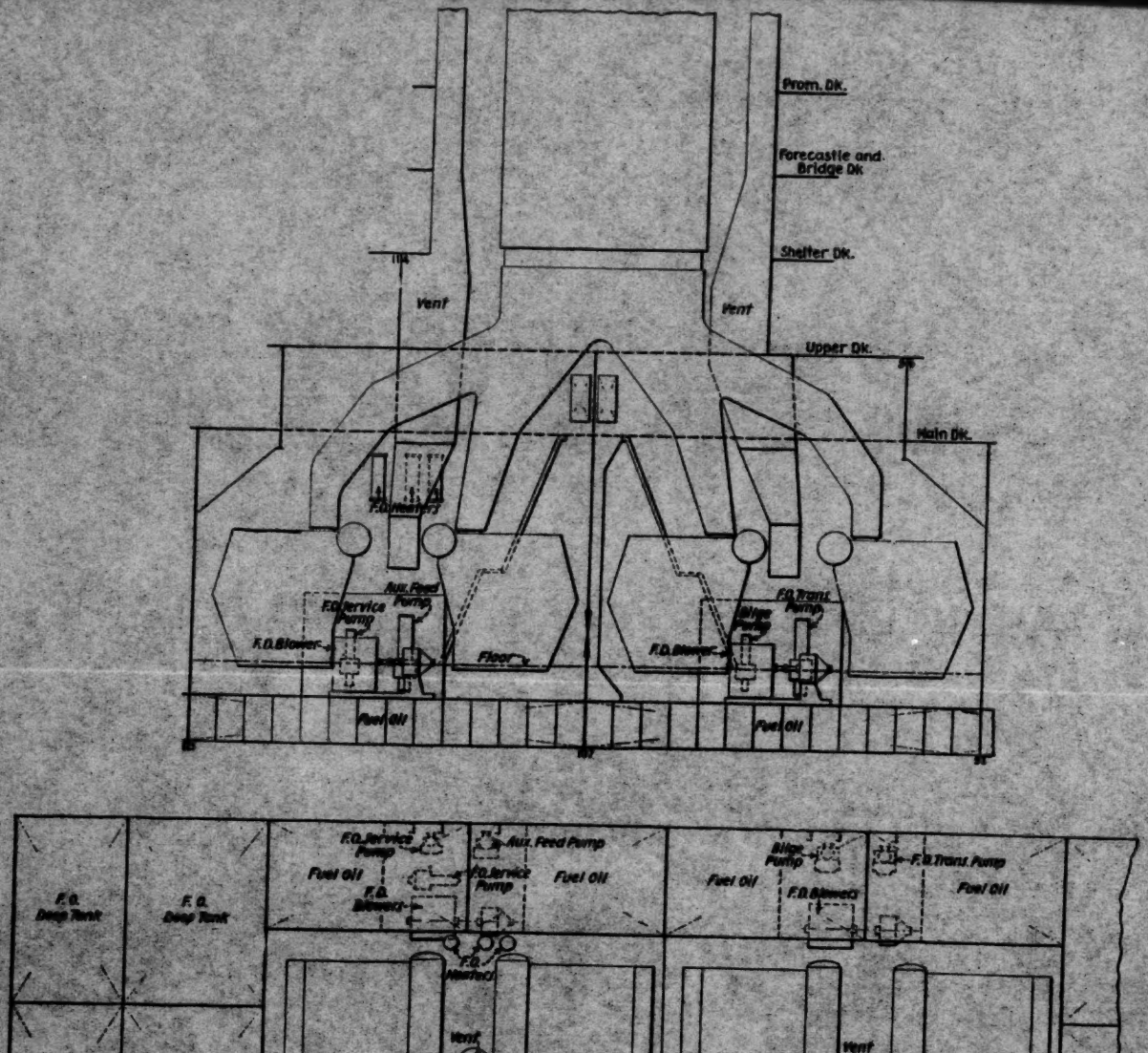
Plan of engine room

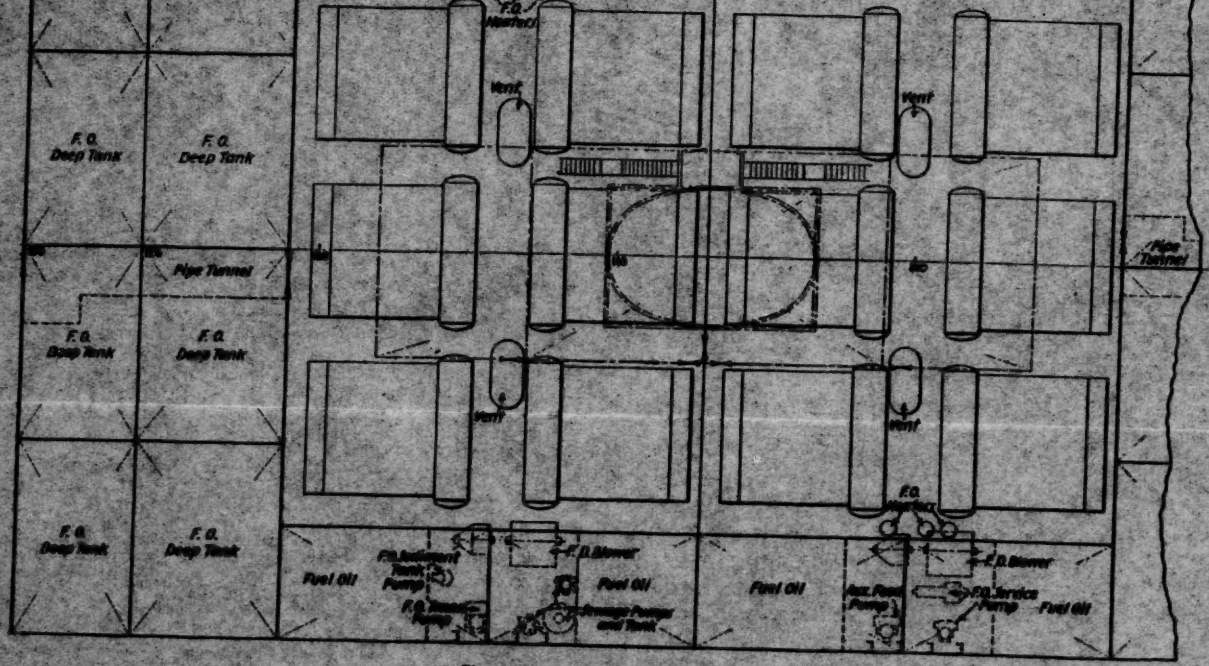
Plan of machinery flat

Plan of engine room

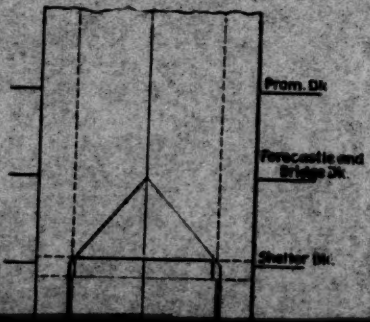


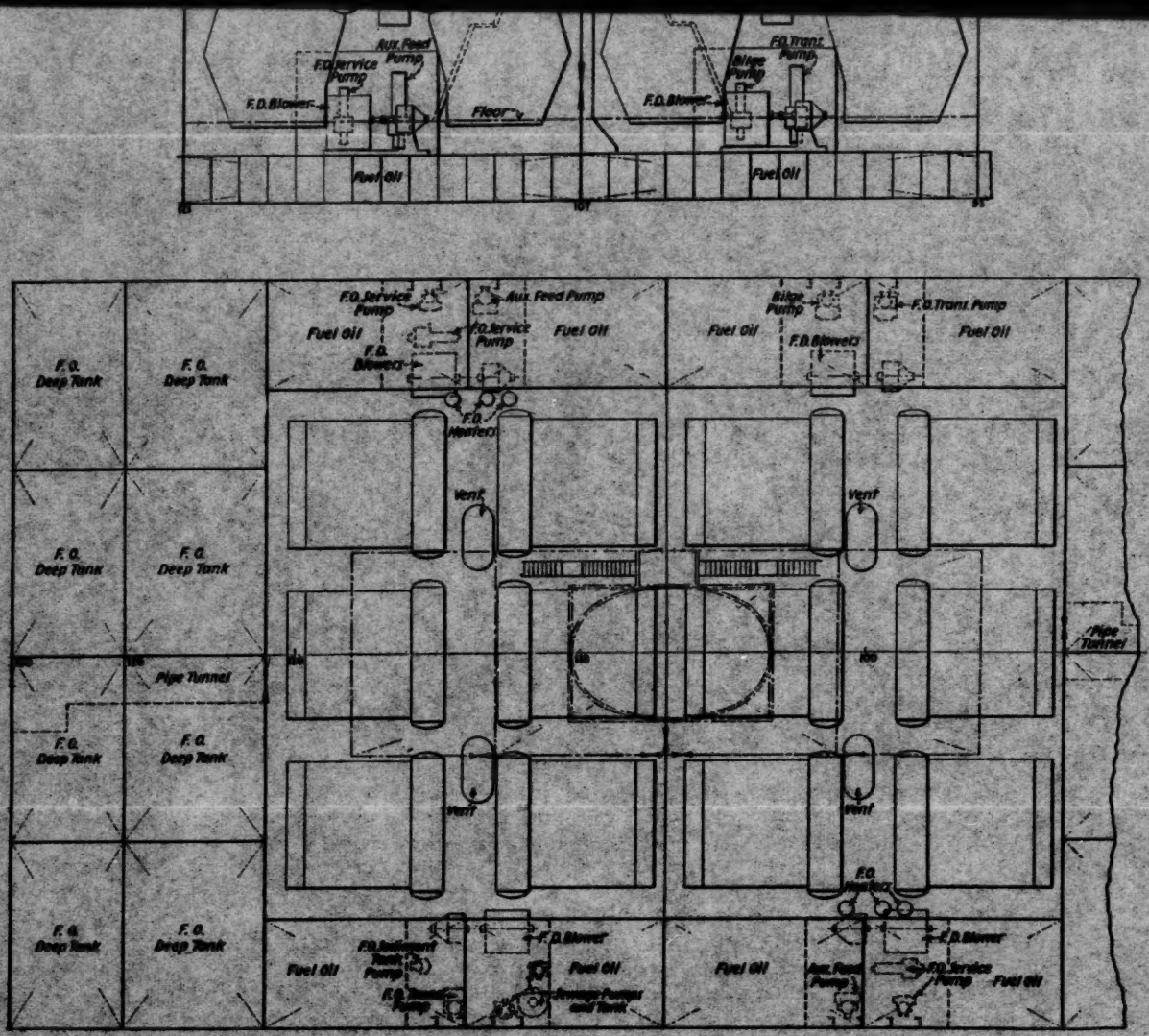
Section through engine room at frame 131, looking aft


Prom. Dk.
Forecastle and Bridge Dk
Shelter Dk.
Upper Dk.
Main Dk.
Vent
Vent
F.O. Heaters
Aux. Feed Pump
F.O. Service Pump
F.O. Blower
Floor
F.O. Trans. Pump
Bilge Pump
F.O. Blower
Fuel Oil
Fuel Oil
F.O. Deep Tank
F.O. Deep Tank
F.O. Service Pump
Aux. Feed Pump
Fuel Oil
F.O. Blower
F.O. Service Pump
Fuel Oil
F.O. Heaters
Bilge Pump
F.O. Trans. Pump
Fuel Oil
F.O. Blower
Fuel Oil
Vent
Vent

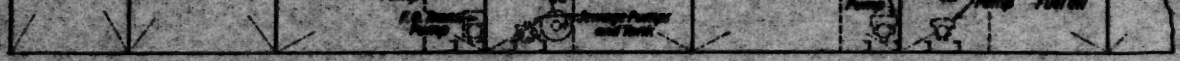


Elevation and plan of boiler rooms

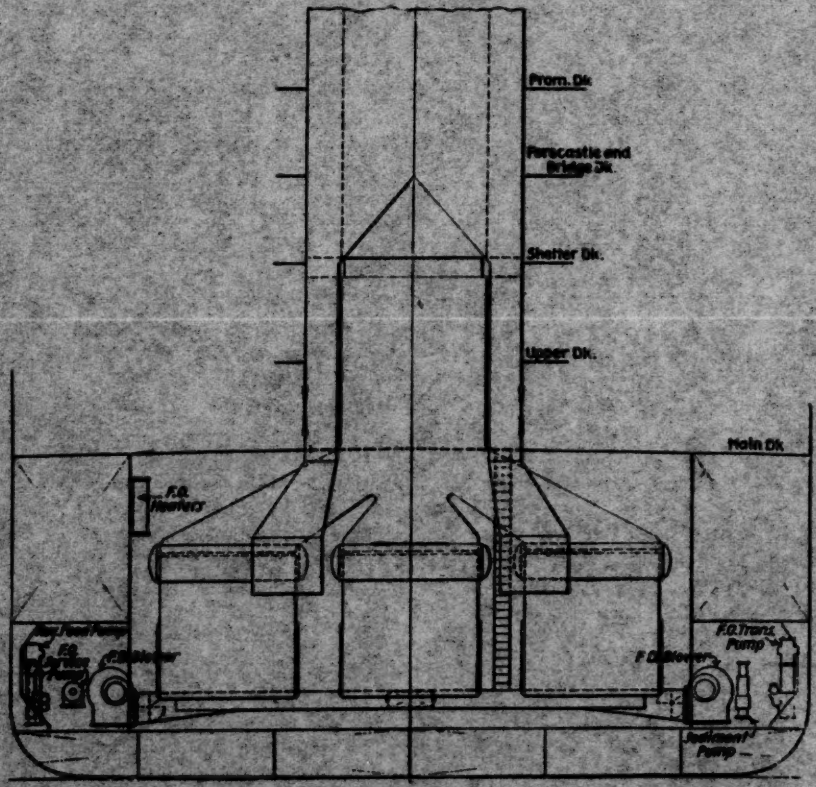




Elevation and plan of boiler rooms



Elevation and plan of boiler rooms



Section through boiler room at frame 121, looking forward

The eight-hour economy run was started at 4.00 P.M. and ended at midnight. It was highly successful in every particular and the data obtained is given in the table.

The following morning, June 19, the port generator and the after fireroom were secured and the vessel was run for several hours on one generator and six boilers at about 16.5 knots. The fuel economy under this condition was about 0.78 pound per hour per shaft horsepower. Following the conclusion of this trial, the customary anchor windlass and steering trials were satisfactorily completed.

The *President Coolidge* was not given her sea trials until September. She had no standardization trial, the data obtained in the case of the *President Hoover* being considered applicable to both vessels.

Early on the morning of September 10, the *President Coolidge* left Newport News and proceeded to sea. Off the Virginia Capes, the compasses were adjusted and the anchor windlass and steering trials were successfully conducted. Upon conclusion of these trials the vessel headed out to sea for her eight-hour economy trial, which was equally as successful as that of the *President Hoover,* and the data obtained is embodied in the same table with that of the latter vessel.

A brief trial of several hours duration with one generator and six boilers at about 18 knots was also conducted following the eight-hour trial. The fuel economy for this trial was about 0.74 pound per shaft horsepower.

STANDARDIZATION TRIAL DATA—S. S. "PRESIDENT HOOVER."

Run	*Speed—Knots*	*R.P.M.*	*Portable K.W. Indicator S.H.P.*
1	16.341	94.41	8,746
2	14.772	94.99	9,140
3	16.172	95.01	8,690
Avg.	15.514	94.85	8,929
5	17.630	106.72	12,223
6	16.822	106.25	12,778
7	17.442	106.5	12,218
Avg.	17,179	106.43	12,499

8	18.302	114.84	16,329
9	18.339	115.47	15,398
10	18.663	114.68	15,930
Avg.	18.411	115.12	15,764
11	19.355	123.81	19,517
12	19.780	123.21	19,704
13	19.608	124.97	20,355
Avg.	19.631	123.80	19,820
14	20.560	129.89	23,691
15	20.443	131.74	24,236
16	20.571	132.30	24,838
Avg.	20.504	131.42	24,250
17	21.480	139.23	29,394
18	21.277	139.58	30,050
19	21.792	141.50	30,844
20	21.327	143.49	32,381
21	22.195	143.36	32,537
Avg.	21.558	141.48	31,060

8-HR. SEA TRIAL DATA.

	Hoover	*Coolidge*
Speed in knots	20.83	*21.2
Slip of propellers per cent mean	12.8	12.6
S.H.P., total both shafts	26,495	27,089
Draft, mean on trial, feet and inches	25 11	24 7
Displacement corresponding tons	24,330	22,927
Pressures (average):		
Main st., at boilers, pounds gage	289	294
at throttle, pounds gage	272	276
at 1st stage nozzles, pounds abs.	138.5	163
Barometer, inches of mercury	30.0	30.1
Vacuum, main condensers referred to 30" barometer	28.80	28.58
Main feed, at pumps, pounds gage	354	372
Main feet, at boilers, pounds gage	328	317
Auxiliary exhaust, pounds gage	7	7.2
Forced lubrication system, pounds gage	15.6	16
Air presure to boilers, inches of water	0.94	0.96

* From *Hoover* standardization with allowance for difference in draft.

Main generators, volts	4,970	4,099
amperes	1,168	1,425
field volts	121	121
field amperes	320	508
R.P.M.	2,688	2,729
Main motors, amperes	1,168	1,425
R.P.M.	134.4	136.43
Temperatures, degrees F. (average) .		
Main injection	55	74
Main overboard discharge	73.8	84
Feed water	312	305
Main steam at superheater outlet	616	.....
Main steam at main turbines	608	626
Engine room, working platform	92	106
Firerooms, working levels	83	98
Outside air	65	78
Smoke pipes	430	424
Main generator coils	126	169
Main motor coils	148	144
Per cent CO_2 (average)	13.2	13
Fuel consumption, pounds per hour, corr'd to 18,500 Btu. oil	17,720	19,094
pounds per s.h.p.-hr.	.669	.705

In conclusion I wish to acknowledge the courteous co-operation of the builders, The Newport News Shipbuilding and Dry Dock Company, who furnished all information and data for the preparation of this article, including drawings and photographs.

NOTES.

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PROPELLING PLANT OF THE KÖNIGSBERG-CLASS OF GERMAN CRUISERS.
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ARMISTICE DAY.

H.M.S. *ACHERON*.

BRITAIN'S FIRST HIGH-PRESSURE WARSHIP. TECHNICAL DESCRIPTION OF MACHINERY WHICH HAS PROVED OUTSTANDINGLY ECONOMICAL.

H.M.S. *Acheron* is one of the "Acasta" class of destroyers, and has a designed speed of about 35 knots at an output of about 34,000 S.H.P.; her displacement is 1330 tons, with a length of 312 feet between perpendiculars. Her armament comprises four 4.7-inch guns, two 2-pounder pom-poms, five machine guns, and eight 21-inch torpedo tubes on quadruple mounts. The contract for her construction was placed by the British Admiralty with The Parsons Marine Steam Turbine Co. Ltd., Wallsend-on-Tyne, who constructed the turbine machinery and fitted the complete installation on board the vessel, the hull and boilers being built by John I. Thornycroft & Co. Ltd., at Woolston, Southampton.

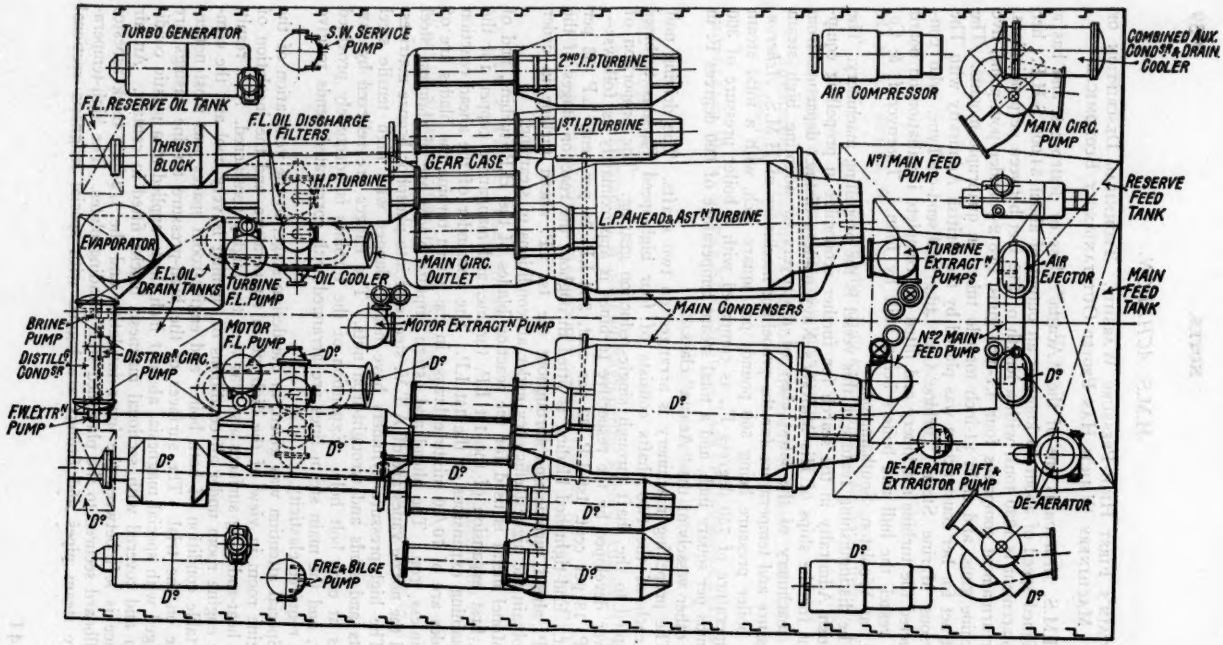
The distinguishing feature of this vessel is the propelling machinery. The British Admiralty in the search for further economies in propelling equipment for the ships of His Majesty's Navy made a distinct departure from the machinery of the latest class of destroyers by adopting high steam pressure and temperature for the propelling machinery of H.M.S. *Acheron*, the boiler pressure being 500 pounds per square inch, with a total steam temperature of 750 degrees F., as compared with a boiler pressure of 300 pounds per square inch, and a total steam temperature of 600 degrees F. in the other vessels of the "Acasta" class.

The propelling machinery is arranged on two shafts. The driving unit of each of the two shafts consists of four high-speed Parsons turbines coupled to the shaft through single-reduction gearing. The proportion of power developed by the respective turbines is approximately as follows:—H.P. 18 per cent, 1st I.P. 23 per cent, 2nd I.P. 26 per cent, L.P. 33 per cent. End-tightened blading is fitted throughout the reaction stages of the H.P. ahead turbine and throughout the 1st I.P. ahead turbine in order to obtain the highest efficiency under all conditions of steaming.

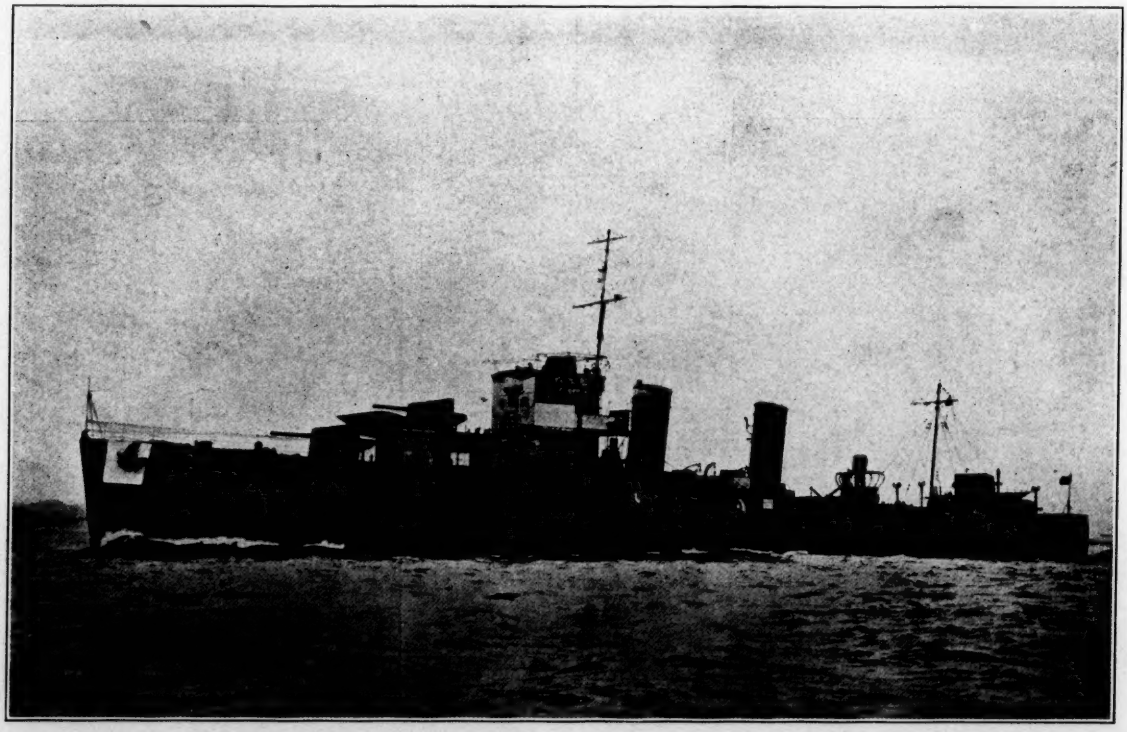
Monel metal is used for the reaction blades of the H.P. turbine, and of the first expansion of the 1st I.P. turbine, and manganese copper for the remaining expansions of the 1st I.P. The remainder of the ahead reaction blades are of 70/30 mixture brass; in the astern turbine the blades are of stainless iron. The ahead and astern impulse blades are of stainless steel and the nozzle vanes are of A.T.V. steel cast into phosphor bronze matrices.

The high-pressure cylinders have heavy flanges with high tensile steel bolts and studs and wrought-iron nuts. The flanges are relieved by saw cuts at each bolt hole. Expansion of the turbines is adequately provided for, and the main steam lines are arranged with corrugated bends to give the required elasticity.

Special attention was given, in the design stage, to the ventilation of the engine room in view of the high steam temperature, and the insulation of the heat-radiating surfaces also was very thoroughly considered. As a result, the engine room under all conditions of running is very cool and the comfortable condition of both boiler and engine-room spaces was most noticeable on the trial. The surfaces of the high-pressure turbine casings are lagged with special magnesia, about 1 inch thick, applied in a plastic condition and covered with sectional magnesia blocks fitted in contact. An air space is left between the inner insulation and the 1/2-inch thick asbestos millboard secured to the planished steel cleading plates. The high-temperature steam pipes and fittings are lagged with special magnesia sectional

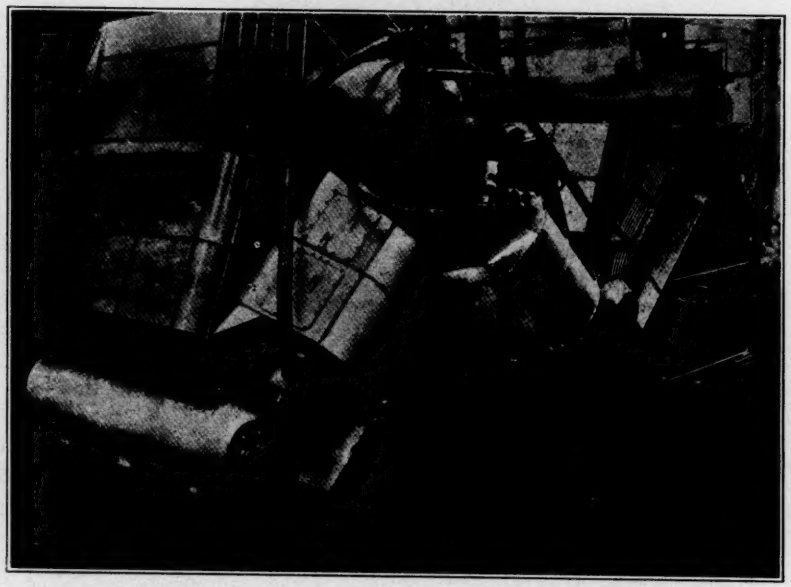


General Arrangement of the Engine Room of the High-Pressure Torpedo-Boat Destroyer "Acheron."

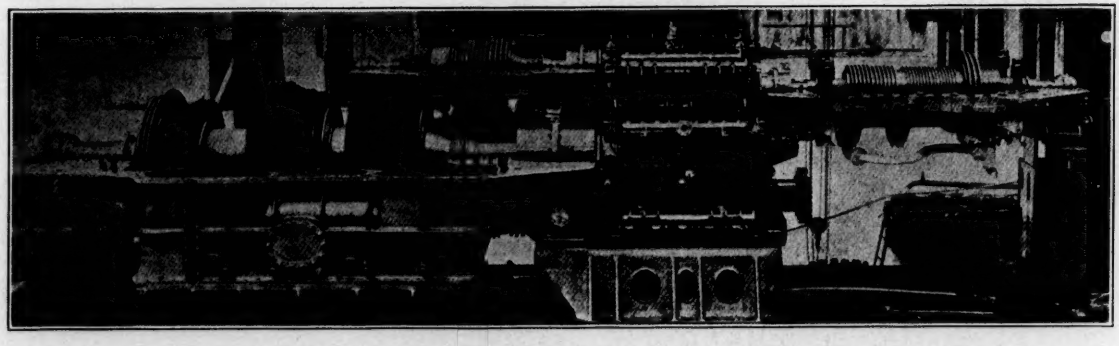


H.M.S. "Acheron"—Britain's First Warship to be Propelled by High-pressure, High-temperature Steam Machinery. The "Acheron" was Built and Boilered by John I. Thornycroft & Co. Ltd., and the Propelling Machinery was Supplied by The Parsons Marine Steam Turbine Co. Ltd.

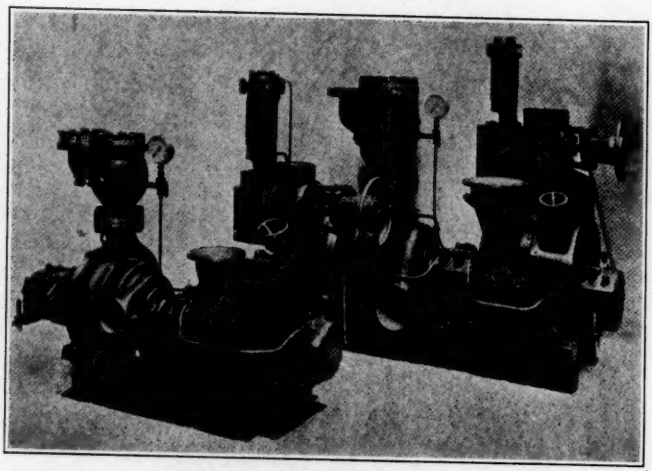
One Set of Turbines, Showing the Double-Flow L.P. Turbine in the Foreground.



Two of the Thornycroft Boilers, Which Have Admiralty Type Superheaters.



A Side View of Set of Main Turbines and Gearing for the "Acheron" with Covers Removed.



WEIR TURBO-FEED PUMPS.

blocks between covers of asbestos cloth. The joint flanges are fitted with magnesia mattresses specially arranged for easy removal and examination of the pipe joints when desired.

All the auxiliary machinery is arranged to work with steam at a reduced pressure of 300 pounds per square inch and to exhaust against a back pressure of 35 pounds per square inch. This arrangement involved very little departure from the established practice of the makers of this class of machinery. The reduction in the steam pressure is obtained partly by use of reducing valves and partly by restriction in the sizes of the auxiliary steam pipes and connections.

Steam enters the H.P. ahead turbines through nozzles which are specially grouped for use under low power and cruising conditions. Under full power conditions, H.P. steam is admitted direct to the reaction portion of the H.P. turbine, but a flow of steam is maintained through the impulse nozzles of the cruising element under all powers.

Labyrinth packing is fitted in the glands arranged at both ends of each turbine, with suitable pockets and connections for the control of the supply and leak-off steam as required for sealing purposes. The screwed column type of lifting gear is supplied for all turbines and permits ready examination of the rotors, blading, &c. Suitable doors are arranged on the gearcases for examination of the pinion and gearwheel teeth.

The gearing is of the double-helical type, the wheel being built up of steel plates, with loose rims bolted on, in accordance with usual naval practice. The main thrust block is of the single collar self-adjusting, pivoted type.

The condensers are connected directly to and underneath the L.P. casing and arranged with rolling chocks to obviate distortion of the cylinder casing when the vessel is rolling heavily. Rolled steel plate and angle construction is used for the condensers, which are of the single-flow type. The circulating water inlets and discharges are arranged to give a scoop effect which is assisted, as required, by turbo-driven geared circulating pumps. This arrangement was found to work very satisfactorily throughout all the trials. The port and starboard propelling units are entirely separate, and may be independently operated as desired. The maneuvering valves are arranged at the forward end of the L.P. turbine and the necessary pressure gage boards, telegraphs and other instruments are conveniently placed in clear view of the engineer on watch.

Forced lubrication pumps—one turbo-driven, working, and one standby motor-driven—are fitted adjacent to the two oil drain tanks, which are arranged at the after end of the engine room in order to obtain the maximum fall from the turbines and gearing. All turbine and gearing bearings, the main thrust blocks and the gearing sprayers, are lubricated on the forced pressure system. Sight flow indicators are fitted in the drain pipes from each bearing. Oil coolers are provided to maintain a constant temperature of oil in the system and strainers in the discharge pipes catch any foreign material before re-entering the system. An oil separator is fitted to cleanse the oil of dirt or water.

Water-extraction pumps and two-stage air ejectors draw the water and air from the condensers. A well is provided in the base of each condenser to provide a reserve supply for the water-extraction pump when no steam or make-up water is entering the condenser. The level of water in this well is regulated by float-controlled overflow and make-up feed valves having connections to or from the main feed tank. A hand regulated make-up feed connection is also provided from the reserve fresh-water tank. The

boiler feed connections are arranged on Weir's closed feed system; the water-extraction pumps deliver through the air ejector coolers to the Weir main feed pumps, thence through feed heaters to the feed-water regulators on the boiler steam drums.

An emergency turbo-driven feed pump for each boiler is arranged to draw from the reserve feed tank, and the pumps are kept primed and ready for instant operation by a small motor-driven air extraction pump fitted in each boiler room.

For harbor purposes, the exhaust steam from the auxiliaries in use is led to a combined drain cooler and auxiliary condenser which is circulated by sea water. The drainage therefrom passes through a filter and thence to the main feed tank. A de-aerator system is provided and comprises a de-aerator, a motor-driven combined extraction and lift pump and a separate turbo-driven harbor service feed pump. Water drawn from the main or reserve fresh-water tank is freed from its air content in the de-aerator and discharged by the extraction pump direct to the harbor service feed pump.

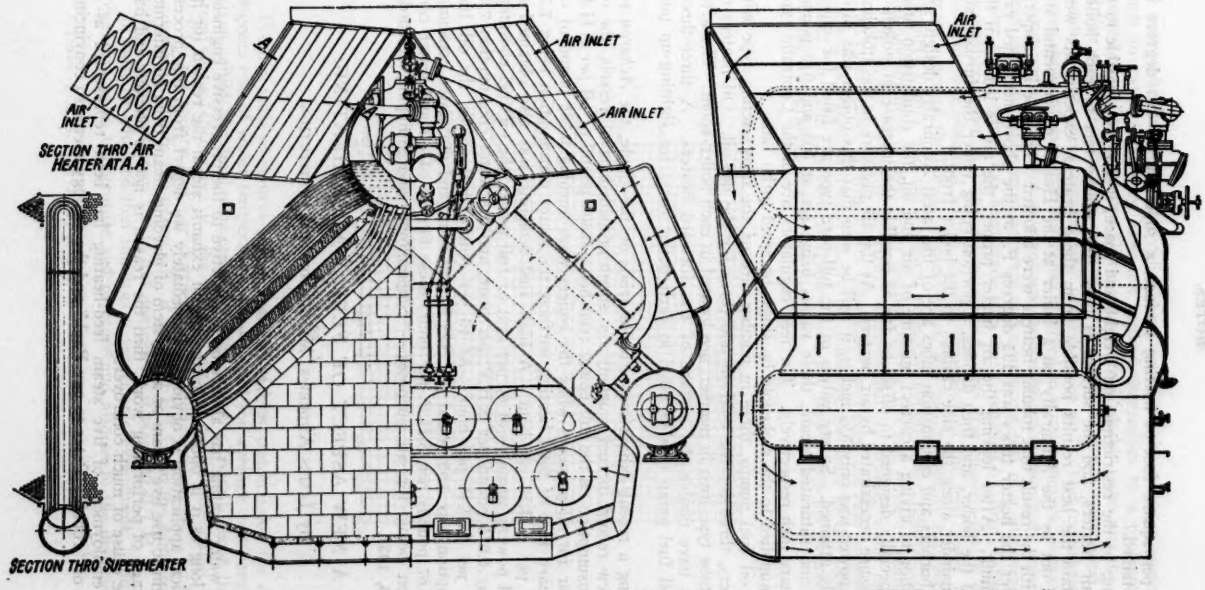
Two turbo-driven generators are fitted in the engine room, each having an output of 80 Kw. at 230 volts. A 20-Kw. oil engine-driven generator is fitted at the forward end of the vessel for harbor and general use as required. A 50-ton capacity evaporating and distilling plant is fitted in the engine room, having connections for using both main boiler steam and exhaust steam from the auxliary range. The rotary type pumps are motor-driven. An electro-hydraulic steering unit is fitted in a compartment directly above the rudder.

BOILER PLANT.

The boiler rooms, two in number, are arranged on the closed stokehold system, air being supplied by six forced draught fans, which are driven by geared turbines. The forward room has one boiler and the aft room two boilers, as usually arranged. The three boilers are of the three-drum small-tube type, designed by John I. Thornycroft & Co. Ltd., in collaboration with the Admiralty, and the boilers, superheaters, and air heaters were manufactured in the Woolston works. The boiler drums are of forged steel, with ends closed in and machined internally and externally. The generating tubes are of solid drawn steel, the two fire rows being 1¾ inches, next four before superheaters 1½ inches, and the remainder 1⅛ inches external diameter.

To each boiler, superheaters are fitted, of the two-flow hair-pin type, with solid forged cylindrical headers. The ends of the headers are solid with the barrels, and to avoid fitting riveted pads, the drums are thickened in way of the boiler mountings. The tubes are 1⅛ inches external diameter, and of low carbon steel. They lie horizontally in the nest of generating tubes, the headers being parallel to the front of the boiler. This type of superheater is as used in practically all Admiralty vessels fitted with superheat boilers, and is similar to that adopted as far back as 1914, in the warships *Meteor* and *Mastiff*, designed and built by Thornycrofts. These earlier designs were not, however, so ambitious in respect of degree of superheat and utility over extended ranges of power, and in order to obtain improvements in these respects a series of experiments was carried out by the Admiralty, with the result that the superheat can now be maintained over a wide range of boiler rating. The results of these experiments have been embodied in the present boilers.

The *Acheron's* boilers were designed for a superheat of 730 degrees to 750 degrees F., at full power, and when burning oil at the rate of 0.785



SECTION AND ELEVATION OF ONE OF THE THORNYCROFT WATER-TUBE BOILERS OF H.M.S. "ACHERON."

pound per square foot of heating surface, a superheat of 725 degrees F. was obtained.

Owing to the restrictions on weight and space, it was not feasible to fit large air heaters, and the surface is only 1650 square feet to each boiler. To obtain the best results possible from these heaters, experiments were carried out by the Admiralty on a boiler at the Haslar experimental station, with the result that tubular heaters were adopted. The rise in temperature over the heater tubes was 135 degrees F., at the above-mentioned rate of burning. After leaving the air heater proper, the air is led down the back of the boiler, and then divides, a portion going under the furnace and the remainder along the side casings to the boiler front.

The burners and combustion tubes are of the latest Admiralty design, and in addition to giving a correct mixture of air and oil, the combustion tubes were specially designed to offset as far as possible the increase in stokehold air pressure necessitated by air heaters. As the air pressure in the stokehold at full power was only 3½ inches, it will be seen that this object was satisfactorily attained. Safety valves of the full-bore type are fitted for superheated and saturated steam, these valves being set at 505 and 515 pounds per square inch respectively. The feed regulators are of the Contraflo type. Air is supplied to the stokehold by single-breasted turbo fans.

The oil fuel pumps, three in number, are of the direct-acting type with six filters, three for the cold oil and three for the hot oil. Oil heaters of the U-tube type, two in number, are located in each boiler room. The after heaters have double the surface of the forward heaters. A three-throw hand oil fuel pump is also fitted in each boiler room for lighting-up purposes.

During a recent exhaustive series of trials on the Clyde, the *Acheron* set up a new record in fuel economy for steam-propelled naval vessels, the oil fuel consumption on the full power trial being 0.608 pound of oil per S.H.P. per hour for all purposes when the boilers were burning 0.785 pound of oil per square foot of total heating surface, the water consumption being 7.77 pounds per S.H.P. per hour. A high fuel economy was also obtained at reduced powers, namely—at 60 per cent of full power, the oil fuel consumption was 0.675 pound per S.H.P. per hour, and the water consumption 7.93 pounds per S.H.P. per hour. At 10 per cent of full power, the oil fuel consumption was 0.92 pound per S.H.P. per hour, and the water consumption 12.87 pounds per S.H.P. per hour, these figures of oil and water consumption being for all purposes.—"Shipbuilding and Shipping Record," June 25, 1931.

A NEW ASPECT OF FEED WATER HEATING.

By James Andrews, Member of the Institution.

21st October, 1930.

Feed water heating has always been attractive to the progressive engineer, and so long as it is obtained by the use of exhaust steam the reason for its economical application is obvious. Immediately we depart from this process and begin to use live steam or any system of bleeding steam from an intermediate stage of performing work, then the reason is not so obvious and has been the cause of much controversy.

The employment of live steam feed-heating has been represented as a process of "robbing Peter to pay Paul," and the analogy seems so convincing

at first sight that this simple ridicule has perverted the minds of engineers to such an extent that they have not applied themselves to a problem which seemed so obviously against all reason. Even Kirkaldy, who was the pioneer of live steam feed-heating, said that "in theory it is not economical, but in practice it has been conclusively proved to be so." This contradiction in terms shows at once that the theory referred to was incomplete; it was in fact wrong, and the proof of its error was demonstrated in numerous vessels from 1883 onwards. The most accurate and convincing test of a live steam feed-water heater which has been made was that conducted by Prof. Gibson on a locomotive boiler in Dundee University.* This test showed that 8 per cent economy was obtained with a cascade discharge of the feed water into the steam space of the boiler. Nevertheless, this incompatibility between an incomplete theory and the practical economy obtained has retarded progress and development in feed water heating during the last 40 or 50 years.

Box, in his "Treatise on Heat," showed that by heating water contained in an open vessel by steam in a lap-welded iron pipe, the rate of heat transmission increased with each rise of the initial temperature of the water to be heated; and the rate of heat transmission per square foot per degree difference of temperature is given in Figure 1.

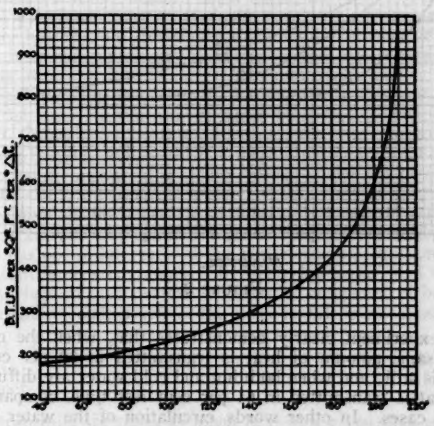


FIGURE 1.

From a table of figures giving the rate of heat transmission for every 10 degrees rise of water temperature, he obtained the average rate between any given initial and final temperature of the water. Thus, to heat the water from 40 to 140 degrees F., the average rate of heat transmission was 266 B.T.U. per square foot per degree difference of temperature. To heat the water from 100 to 200 degrees F., it was 350 B.T.U. on the same basis, and to heat the water from 200 to 212 degrees F., it was 800 B.T.U.

It must be admitted that this was a very crude experiment, limited in its operation as to temperature and pressure and also as to the material of the

* Trans. I. E. S., vol. LIV, p. 241.

heating surface, but it has been confirmed on many practical occasions and it is generally accepted that as water becomes hotter it becomes more lively and diffused, and thereby absorbs heat more rapidly than it does in the cold state.

The author made a simple experiment 30 years ago to show the influence of the diffusion of water on the rate of transmitting heat. This was published in the discussion of Prof. Gibson's paper already referred to, but the result is given here in Figure 2 for a double purpose. A vessel of cold water had a small vessel placed in the middle of it, filled to the same level with hot water. The time taken to cool the hot water from 197 to 80 degrees F. was noted under the following conditions:—(*a*) hot and cold still water, (*b*) hot water still and cold water stirred round or diffused, and (*c*) both hot and cold water diffused.

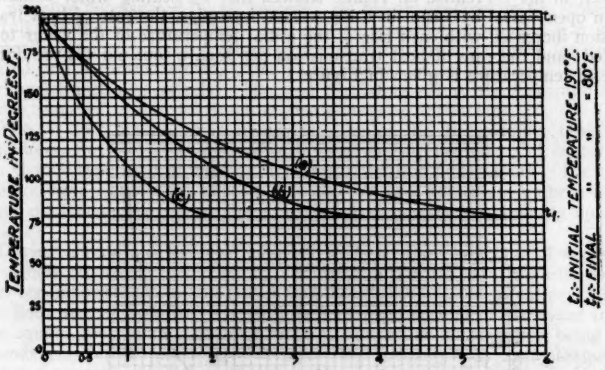


FIGURE 2.

Now this experiment clearly demonstrates that, when the cold water is diffused, the same amount of heat is transmitted in 30 per cent less time than when it is still; and when both hot and cold water are diffused the same amount of heat is transmitted in 64 per cent less time, compared with still water in both cases. In other words, circulation of the water increases the rate of heat transmission, or, to put it in a more scientific form, Prof. Osborne Reynolds' law states—"the amount of heat carried off by any fluid from a surface is proportional to the internal diffusion of the fluid at and near the surface." It will also be observed that this law agrees with Box's finding that heat is transmitted more rapidly at the higher than at the lower temperatures, because the curves fall more quickly at the top than at the bottom.

It has been shown that* in a locomotive boiler 10 per cent of the heating surface generates 65 per cent of the total steam, so that the remaining 90 per cent of the heating surface must be relatively inefficient. In a marine Scotch boiler the proportion of inefficient heating surface cannot be much less in normal circumstances. Now if this large proportion of inefficient

* "The Heat Efficiency of Steam Boilers," by Bryan & Donkin.

heating surface can be rendered more efficient by supplying hot feed-water to the boiler, so that the water will be more lively and increase the rate of heat transmission, then the heating surface remote from the combustion chamber will become more effective, and will extract more heat from the products of combustion; in short, the efficiency of the boiler will be increased.

It has been shown that the rate of heat transmission is greater at high temperatures than it is at low temperatures. It therefore seems obvious that the height of the temperature must be in relation to the boiling point, because the water reaches its most mobile state at that point. If, therefore, feed heating is limited to a temperature of 220 degrees F. for a boiler pressure of 200 pounds per square inch, and a temperature of 387 degrees F., as it usually is at present, it would appear that it is being stopped at a point at which the rate of heat transmission is increasing more rapidly with each increment of feed temperature. But the question which commonly arises is: What advantage is to be gained by heating the feed water above 200 degrees F. with live steam when the boiler is there for that purpose? We know that economy in coal consumption and other well-defined benefits arise due to feed heating by exhaust steam up to 200 degrees F., but it has not hitherto seemed so reasonable to exceed that limit by the use of live steam.

That question brings us to the relative rate of heat transmission in a boiler and in a feed water heater. In a boiler the heat is transmitted from a dry gas to water, which is a relatively slow process compared with that in a feed water heater from steam to water. In the boiler a dry gas is a bad conductor of heat compared with steam in a feed water heater, because as soon as steam enters a feed heater it becomes vapor and a film of water covers the whole heating surface. In the latter case the heat is first transmitted from the vapor to the film of water by direct contact, then from the film of water through the heating surface to the feed water. As Rankine on the "Steam Engine" puts it—"The most rapid convection of heat is that which is effected by means of a cloudy vapor, which combines the mobility of a gas with the comparatively greater conducting power of a liquid as when steam communicates heat to a solid body by condensing on its surface."

But there is another important difference between the boiler and the feed heater which has not so far received due consideration. In general, the heating surface in steam boilers is composed of steel plates and steel tubes, whereas the heating surface in feed heaters is generally composed of brass or copper tubes. The relative conductivity of the three metals is as follows:—

Steel	15.5
Brass	22
Copper	92

If copper transmits six times as much heat in a given time as steel, why is copper not used for the heating surface of boilers? In the first place it was discovered that when a copper heating surface has acquired a coating of soot its conductivity is little better than iron or steel with a similar coating of soot. Secondly, copper is not so strong or reliable at the working temperature in a boiler. It has been successfully used in locomotive fireboxes, but there it is kept free of soot by the intense direct heat of the incandescent gases. There is not such a serious heat-retarding influence in a feed water heater. Sometimes a film of oil accumulates on the tubes but this is soon revealed by the temperature of the feed water, and means are generally provided for cleaning this off easily when desired.

In order to show definitely that the rate of heat transmission increases with an increase in the temperature of the water, the apparatus shown in Figure 3 was constructed. The water container was open to the atmosphere and was

fitted with vertical copper tubes, and a ring gas-burner giving a temperature of 900 degrees F. was situated in the space below the container. Numerous tests were made, beginning with a known weight of water at 50 degrees F. and taking readings every 30 seconds, and when evaporation was seen to take place the amount evaporated was weighed at three of the higher temperatures to locate the true point on the curve shown in Figure 4. This curve shows the continuous increase in the rate of heat transmission per degree difference of temperature from 162 degrees F., below the boiling point up to the boiling point. In Blechynden's experiments on the transmission of heat through plates,* the water was maintained at a constant temperature while the gas temperature was increased, and those results also show that the rate of heat transmission was increased with every increment of increase in gas temperature.

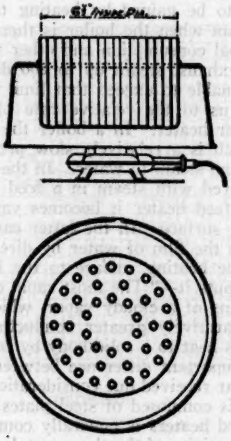


FIGURE 3.

Both of these experiments were made at atmospheric pressure and with comparatively low gas temperatures, and they should not be taken as representing what actually takes place in a boiler, because a boiler is being constantly supplied with water much below the boiling point, and the range of temperature between the feed water and the steam is much greater; consequently, there would be more scope for improvement in a boiler. However, this curve is merely intended to illustrate the point that when transmitting heat from a dry gas to water the rate of heat transmission increases with an increase of the water temperature. The actual rate of heat transmission in a marine boiler varies considerably, but taking a good class of boiler having an efficiency of about 75 per cent, the rate is about 4600 B.T.U. per square foot of heating surface per hour.

A live steam second-stage feed-water heater fitted with copper tubes and having the inlet water temperature at 200 degrees F. will transmit about 46,000 B.T.U. per square foot of surface per hour, or ten times as much

* Trans. I. N. A., 1893.

heat per unit as the boiler transmits. Now suppose a marine boiler to be generating 30,000 pounds of steam per hour from 200 degrees F. to 200 pounds gage pressure, and it is proposed to fit a live steam feed-water heater to raise the feed temperature to 300 degrees F. Originally the amount of heat transmitted by the boiler per pound of steam was

$$1200-(200-32) = 1032 \text{ B.T.U.}$$

Assuming that the boiler continues to transmit the same amount of heat per unit of surface when the feed temperature is increased to 300 degrees F., the amount of steam generated per hour will be $\frac{30{,}000 \times 1032}{932} = 33{,}220$ pounds and the additional 3220 pounds of steam is exactly the amount required by the heater to raise the temperature of the water from 200 to 300 degrees F.

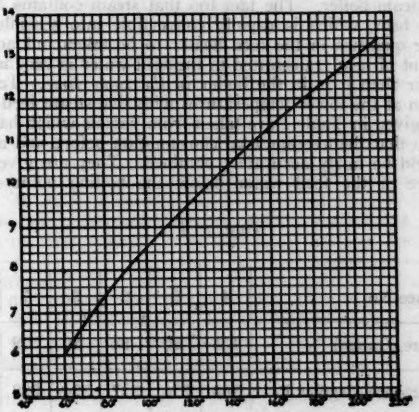


FIGURE 4.

Now we come to the missing link in the theory of live steam feed-water heating. The water now enters the boiler at 100 degrees F. higher temperature, and according to the curve given in Figure 4 the rate of heat transmission has been considerably increased. The precise amount of the increase cannot be given, because no experiments have ever been made to form a curve corresponding to Figure 4 which would be applicable to a boiler under pressure and at a much higher temperature than that at which the curve was made; but there have been a great many trials made with live steam feed-heaters on board ship which give a fair indication of the actual economy obtainable at the lower end of the curve. There have also been many careful trials made to determine the amount of steam obtainable at the top of the curve. It will now be easily understood that the most important consideration in determining the gain to be expected is not so much how high the feed temperature has been raised, but how far it is below the boiling point or steam temperature. It will also be clear that so far as the rate of heat trans-

mission in the boiler is concerned, it does not matter where the heat comes from to heat the feed water—the gain in heat transmission will be the same. If it comes from exhaust steam there will be a double gain, because the heat in the steam will be recovered and the rate of heat transmission in the boiler will be increased. If it comes from steam bled during the process of performing work, provided that the quality of the steam left to complete the work is not unduly reduced, there will be a double gain to a less extent than with exhaust steam. If it comes from an economizer it is also a double gain. If it is taken from the boiler, then the steam used in the heater has to be deducted from the total evaporation.

With the foregoing explanation, it will now be understood that live steam feed-water heating is not merely a process of taking heat out of the boiler to put it back again. It is a process by which a higher rate of heat exchange is effected between steam and water in a feed water heater, than between a dry gas and water in a steam boiler; and also by reason of the superior conductivity of the copper heating surface compared with the steel heating surface in a steam boiler. The fact too that steam contains a large proportion of latent heat, which is rapidly convertible into sensible heat, permits those superior qualities in the feed heater to be taken full advantage of, so that the amount of heat transmitted to the feed water is very much greater in the feed heater than it is in the boiler *in the same time*. To show the approximate gain at the lower temperatures with live steam feed-water heating, the examples given in Table I are taken from vessels which had originally no feed heater, so that the average coal consumption before and after the heater was fitted could be fairly accurately obtained. They are given in the order of temperature of the feed water below the boiling point or steam temperature.

TABLE I

Reference No.	4	6	2	3	7	8	5
Steam pressure (Gauge) - -	155	80	84	80	78	87	167
Steam temperature, degrees F. -	365	324	327	324	322	329	373
Feed nlet temperature, degrees F. -	120	133	128	134	135	138	190
Feed outlet temperature, degrees F. -	220	187	192	200	198	211	300
Feed increase temperature, degrees F. -	100	54	64	66	63	73	110
Feed below boiling point, degrees F. -	145	137	135	124	124	118	73
Coal saved, per cent. - - -	7·4	5·2	8·8	9·5	6·25	7·8	11·6

Those records were taken from separate vessels in which the conditions were no doubt variable; they were certainly variable as regards the steam pressures and there was a difference in the increase of water temperature, but the general conclusion appears to be that the nearer the feed water temperature approaches the steam temperature, or boiling point, the greater is the percentage of economy. For the purpose of this paper, great accuracy in the percentage of gain is not essential at the lower feed temperatures, because the maximum variation under the same conditions would be very small, whereas the percentage of gain at the higher feed temperatures varies considerably with a relatively small variation of the feed temperature.

There are three ways of obtaining economy; the first is by generating the same amount of steam with less coal burned in the boiler furnace, which is the usual condition as applied to existing marine machinery. The second is by generating more steam with the same boiler, which is the method adopted by London electric power stations. The third, which would be applicable to any new installation of machinery, is to generate the same amount of steam with less boiler heating surface. Prof. Rankine in the "Steam Engine" recognized that a feed water heater should be considered as a part of the boiler, because he says that when computing the efficiency of a boiler the heating surface of the feed-water heater should be added to that of the boiler. The kind of feed heater which Rankine referred to in 1859 was what we now call an economizer and it was fitted in the chimney. If, therefore, the amount of boiler heating surface was dependent upon the amount of heating surface in an economizer in Rankine's time, how much more necessary is it today that the amount of boiler heating surface should be reduced in relation to the much more efficient heating surface in a live steam feed-water heater? In short, when heat has been imparted to the feed water in a feed heater, the boiler has that much less heat to transmit in the generation of steam, and on that account alone the boiler heating surface should be reduced.

Now if the whole of the water in a boiler, including the feed water, could be raised and maintained close to the boiling point or steam temperature, the maximum rate of heat transmission per pound of steam generated, or the maximum rate of steam generated per square foot of heating surface, should be obtainable. Those are precisely the conditions obtainable with Druitt-Halpin's thermal storage tank system. It may be well to explain here that Druitt-Halpin's system of storing heat was invented in 1892 to replace the costly system then in common use of having electric storage batteries to overtake the short period of peak loads in electric power stations. The influence of a high feed temperature on the rate of heat transmission in the boiler was unknown, and consequently it was estimated that four storage tanks would be necessary to provide for a given overload, where one was found to be sufficient in actual practice. Moreover, calculations based on a normal rate of heat transmission with the usual feed temperature of 100 degrees F., indicated a saving in boiler power during the peak load of 22 per cent, whereas this was greatly exceeded. As ultimately put into operation, the Druitt-Halpin system consists of a large tank, very much like an egg-ended boiler, which is fitted with a water gage and filled to a certain water level. During periods of light load on the generating station steam is taken from the boiler and discharged into the water in the tank, so that when the peak load comes on the water has been heated nearly to the steam temperature. The boiler feed pumps are then stopped and the hot water is fed from the storage tank into the boiler through a regulating valve.

At the Wood Lane generating station of the Kensington and Knightsbridge Electricity Co.,* the normal evaporation of the boilers was 12,000 pounds

* "The Engineer," 3rd June, 1910.

of steam per hour at 200 pounds gage pressure and 387.5 degrees F. After the Druitt-Halpin system of feed heating was installed the temperature of the feed water was 384.5 degrees F., or 3 degrees F. below the steam temperature, and the amount of steam generated by the same boilers was 26,080 pounds per hour, an increase of 117 per cent. When evaporating 12,000 pounds of steam per hour the feed water temperature was 100 degrees F., and the total heat at 200 pounds pressure was 1200 B.T.U. Therefore if the rate of heat transmission had been the same with the high feed temperature, the total evaporation would have been:—

$$\frac{12{,}000\ [1200-(100-32)]}{1200-(384.5-32)}=\frac{1200\times1132}{874.5}=16{,}030 \text{ pounds per hour.}$$ But

the actual evaporation was 26,080 pounds, or 62.7 per cent more, so that the actual rate of heat transmission of the higher feed temperature was 62.7 per cent greater than at the lower feed temperature.

At the St. Pancras Borough Council station,* the normal evaporation of the boilers was 11,000 pounds of steam per hour at 185 pounds pressure and 381.5 degrees F. After the Druitt-Halpin system of feed heating was installed, the temperature of the feed water was 360 degrees F., or 21.5 degrees F. below the steam temperature, and the amount of steam generated by the same boilers was 17,542 pounds per hour, an increase of 59.5 per cent. At the normal evaporation of 11,000 pounds of steam per hour the feed temperature was 100 degrees F.; the total heat at 185 pounds pressure was 1198.3 B.T.U., and on the same basis as the Wood Lane test the total evaporation would have been:—

$$\frac{11{,}000\ [1198.3-(100-32)]}{1198.3-(360-32)}=\frac{11{,}000\times1130}{870}=14{,}300 \text{ pounds per hour.}$$ But

the actual evaporation was 17,542 pounds per hour, or 22.6 per cent more, so that the actual rate of heat transmission at the higher temperature in this case was 22.6 per cent greater than at the lower feed temperature.

Since the introduction of the Druitt-Halpin system in 1893 there have been many installations fitted to electric power stations, and all of them have given approximately the same great increase in the amount of steam generated, according to the temperature of the feed water in the storage tank. There can be no doubt about the facts, and when the curves in Figures 1 and 4 are considered, showing the increasing rate of heat transmission as the feed temperature rises, there should be no reason to doubt that somewhat similar savings in boiler power are available to any enterprising shipowner. The case as it affects the shipowner is not the same as that of the London power stations, but it is equally important and does not seem to present any greater difficulties. The very large passenger vessels now being built require enormous boiler installations. If feed water heaters can transmit heat ten times faster than a boiler, it is obvious that the more efficient heater should be used up to the highest limit, and the less efficient heating surface of the boiler should be reduced as far as practicable; it is not more steam that is wanted on board ship, but a definite quantity of steam from less boiler heating surface.

Let us now return to the example of feed heating referred to on page 649, but instead of limiting the feed temperature to 300 degrees F. let us bring this up to 3 degrees below the steam temperature, to make it the same as the Wood Lane boilers of which definite results have been given. Then—

Feed temperature $= 387.5 - 3 = 384.5$ degrees F.

* loc. cit.

Heat supplied by boiler = 1200 — (384.5 — 32) = 487.5 B.T.U. per pound. The percentage of steam required to heat the feed water is

$$\frac{384.5 - 200}{847.5} \times 100 = 21.8.$$

But, the Wood Lane boilers generated 117 per cent more steam when the feed water was heated to 3 degrees below the steam temperature; therefore, if the feed heater takes 21.8 per cent of the steam to heat the feed water, the net gain is 117 — 21.8 = 95.2 per cent, and the heating surface in the boilers can be reduced by $\frac{95.2}{117 + 100} \times 100 = 43.9$ per cent.

The St. Pancras power station operated at 21.5 degrees below the steam temperature; therefore the feed temperature was

$$381.5 - 21.5 = 360 \text{ degrees F.}$$

Heat supplied by boiler = 1198.3 — (360 — 32) = 870.3 B.T.U. per pound.

The percentage of steam to heat the feed water in this case was $\frac{360 - 200}{870.3} \times 100 = 18.3$. But the St. Pancras boilers generated 59.5 per cent more steam when the feed water was heated to 21.5 degrees below the steam temperature; therefore if the feed heater takes 18.3 per cent of the steam to heat the feed water, the net gain is 59.5 — 18.3 = 41.2 per cent, and the heating surface in the boilers can be reduced by

$$\frac{41.2}{59.5 + 100} \times 100 = 25.8 \text{ per cent.}$$

It is not to be taken that those figures represent exaggerated results, because although the water-tube boilers at Wood Lane were rated at 12,000 pounds of steam per hour, they could supply up to 15,000 pounds of steam per hour, but at this rate there was a tendency to prime, and consequently the boilers were rated at 12,000 pounds per hour to be well within their capacity. According to the report, the same care was taken to keep well within the boilers' steaming capacity at the highest rate of steam generation. The St. Pancras boilers were of the same general design and were rated on the same basis.

Returning to Figure 4, assume in the first instance that feed water is supplied at 90 degrees F. when the rate of heat transmission is 8 B.T.U. per square foot per degree difference of temperature. If the feed temperature is ultimately increased to 190 degrees F. the rate of heat transmission has been increased to 12.5 B.T.U., and the gross saving becomes $\frac{12.5 - 8}{8} \times 100 = 56.25$ per cent. But if steam at atmospheric pressure had been used to raise the feed temperature, the percentage of steam required would have been $\frac{100}{967} \times 100 = 10.35$ and the net gain would have been 56.25 — 10.35 = 45.9 per cent. The amount of boiler heating surface can be calculated from Figure 6 having due regard to the temperature at which the feed water can be maintained, and this will depend upon whether superheated

steam is in use, and whether the first section of the heater is used for desuperheating the steam for the auxiliary machinery. It seems a more rational course to desuperheat auxiliary engine steam in the feed heater than to do it in the boiler, because in the heater it is increasing the feed water temperature generally, instead of the water in the boiler locally.

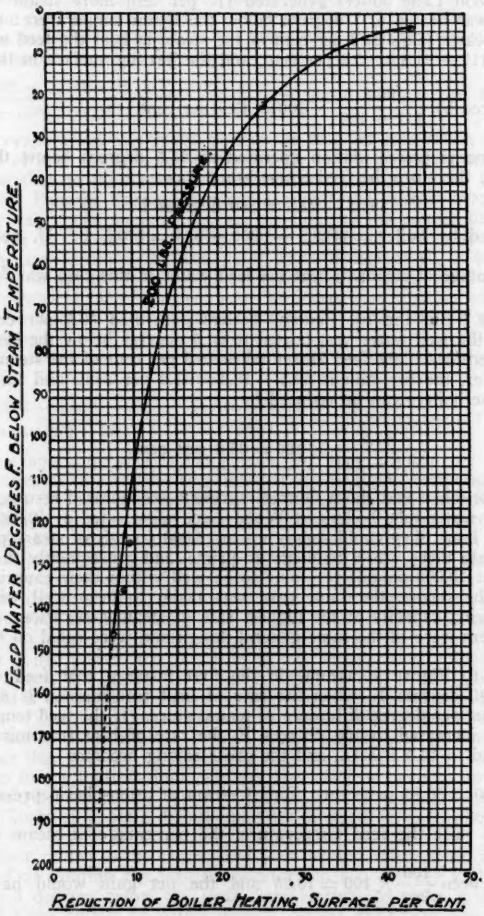


FIGURE 5.

For the complete solution of this problem a boiler should be fitted either with a feed heater or a thermal storage tank on which separate tests could be made with the feed water maintained at fixed increased temperatures, so that by measuring the rate of steam generation, curves could be made some-

what like those in Figures 5 and 6. The lower end of the curve in Figure 5 has been plotted from the results given in Table I, while the Wood Lane and St. Pancras trials give the higher end of the curve. Figure 6 has been calculated from Figure 5 on the assumption that 4.25 pounds of water can

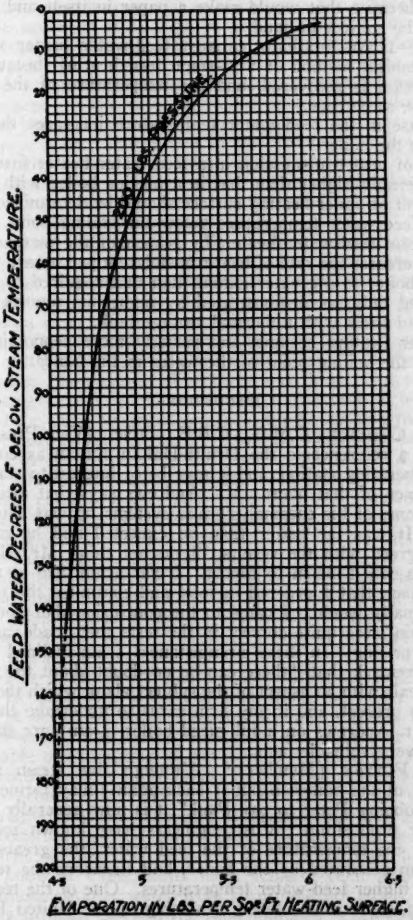


FIGURE 6.

be evaporated per square foot of heating surface per hour, when the feed temperature is 100 degrees F., or 287.5 degrees below the steam temperature. In the absence of more complete trials from a single boiler as indicated above, these curves may be useful as an approximate guide to the determination of

boiler heating surface in relation to the degree of feed heating to be employed. It must be kept in view, however, that the primary subject of this paper is feed water heating, and its influence on the size of the boiler is an obtrusion which the boiler designer is in the best position to solve. In a paper of this character it has not been possible to illustrate any precise design of feed water heater, because that would make a paper in itself and divert the discussion from the subject in hand.

Conclusions.—1. An auxiliary exhaust feed-water heater will always be the most economical method of first-stage feed heating, because the heat in the exhaust steam is recovered, and the temperature of the feed water is increased at the same time.

2. An increase in the feed water temperature increases the rate of heat transmission in the boiler.

3. The rate of heat transmission in a feed water heater fitted with copper tubes is considerably higher than that in a marine boiler with steel tubes.

4. The amount of boiler heating surface should be determined in relation to the degree of feed water temperature below the boiling point.

5. A live steam feed-water heater, by reason of the increased feed water temperature, increases the rate of heat transmission per square foot of heating surface in the boiler to a greater extent than that required to produce steam for feed heating only, as demonstrated by numerous reports of live steam feed-heaters, and trials with a thermal storage tank.

6. Feed water heating becomes increasingly more economical the nearer it is carried to the boiling point of the water in the boiler.

DISCUSSION.

Prof. J. D. Cormack, C.M.G., C.B.E., D.Sc. (Vice-President): This paper contains a reference to the Druitt-Halpin system as employed in the Wood Lane generating station, and it may be of interest to state that I have a little experience of that plant. Mr. Andrews says that after Mr. Halpin had installed some of his thermal storage systems he was astonished at the performance. It was far better than he expected and he called in Prof. Unwin, who agreed with the results. I do not think Mr. Halpin was yet satisfied, for he called me in to report. I was astonished at the results, so astonished, in fact, that I ordered in food and continued the test for double the time I normally would. I came to the conclusion that both Mr. Halpin and Prof. Unwin were quite correct in the tests they made, and the figures really were surprising. In these circumstances I am inclined to agree with Mr. Andrews' remark that "there can be no doubt about the facts."

This paper deals with an aspect of the subject about which there is not very much known at present, and if any firm cares to undertake the experiments indicated by Mr. Andrews on a full-sized boiler, I am sure they could give the Institution very valuable facts.

Mr. W. B. Fulton (Member): Although live steam feed-heating—taking heat out of the boiler to put it back again—may formerly have been regarded as "robbing Peter to pay Paul," it is now generally accepted as a sound economic proposition. Most engineers have known for many years that the higher the temperature of the feed water, the greater will be the amount of steam generated by the boiler, and therefore the tendency today is to adopt still higher feed-water temperatures. One of the technical papers stated the other day that the latest steamship of the United Fruit Co. was fitted with feed heating plant to give a feed temperature of 310 degrees F. This of course is far under the ideal suggested by Mr. Andrews, because the boilers in that ship have a working pressure of 375 pounds per square inch so that the boiler temperature is over 400 degrees F., but it shows that the

tendency is in the direction advocated by Mr. Andrews. Most of us appreciate the fact—although possibly without inquiring into it—that less heat is required to raise the temperature of water from 200 to 300 degrees F. than is required to raise it from 100 to 200 degrees F., just as less heat is required to raise the boiler pressure from 200 to 300 pounds per square inch than is required to raise it from 100 to 200 pounds per square inch.

When trouble arose on trial trips in getting sufficient steam out of the boiler, it was repeatedly found that by fitting an internal feed pipe in the boiler and carrying this round the shell of the boiler once or twice, thus heating the feed water in its course, the steam production was increased to a remarkable degree. If this pipe, instead of once or twice, had been carried perhaps a dozen times round until the temperature of the feed water at the exit from the pipe approximated that of the water in the boiler itself, then probably the results would have been rather surprising. This is a very simple method of heating the feed water, but it has the disadvantage that this piping interferes somewhat with the accessibility of the space round and at the side of the combustion boxes.

If this paper is not misunderstood, Mr. Andrews claims, and substantiates his claim by figures, that if the feed water temperature were raised to that of the boiler water, the steam production would be raised by some 30 per cent, or in other words, that the size of the boiler might be made 30 per cent less in such a case. If this is so, why is it not done? Well, there are two objections. The first of course is the very obvious one that a very elaborate and costly feed heating plant is necessary. In the ship referred to the feed heating plant consisted of three feed heaters in series, the first, it is thought being heated by exhaust steam, the second by bled steam from the turbine, and the third by live steam from the boiler, and that was to raise the temperature only to 310 degrees F. Had the plant been designed to raise the temperature to 400 degrees F.—the same as that of the boiler—it would have been still more costly and elaborate. The second objection is that a large feed heating plant, at a very high temperature, is required in the engine room, and unless this is very thoroughly insulated, especially in the tropics, the lot of the engineer is not a happy one. Neither of these objections is fatal, but some assurance is desirable that this is not again "robbing Peter to pay Paul," and that the saving in the cost of the smaller boiler would not be less than the amount expended on the extra feed-heating plant to obtain this result. I think it would, and I believe that in many cases engineers might, with profit, fit still larger feed heaters even at the expense of a reduction in size of the air heating installation.

Mr. Andrews says that in the Wood Lane station the boiler was rated at 12,000 pounds of steam per hour, and that when the boiler was forced to 15,000 pounds it began to show signs of priming. When the feed water temperature, however, was raised to that of the boiler, or nearly so, the boiler was capable of generating 26,000 pounds of steam per hour. I should like to know whether, under those highly forced conditions, the boiler showed no tendency to prime. Mr. Andrews says further that the transmission of heat is much greater between steam and water, as in the case of a feed heater, than it is between hot gas and water, as in the case of a boiler. Well, no doubt that is true for the reasons given by Mr. Andrews, but it would be interesting to know if experiments have actually been made showing the relative heat transmission to water from steam and hot gas of the same temperature passing through tubes of the same diameter, thickness, and material.

The merit of Mr. Andrews' paper, and it is a great merit, is that it shows rather conclusively, in my opinion, the enormous economy that may be expected from the use of very high feed-water temperatures—temperatures approximating to those of the boiler water itself. This is a matter of much importance and comes to many of us as a revelation.

Dr. OCTAVE SNEEDEN, B.Sc., Ph.D.: Mr. Andrews' paper is of interest in more ways than one: its chief value lies in his attempt to prove the beneficial results arising from feed water heating in general, but I am afraid that his analysis and interpretation of some of the experimental data and tests are open to criticism. Mr. Andrews has viewed the subject almost entirely from the heat transmission aspect, and it is rather remarkable that, with such a wealth of positive information as that offered by the proceedings of this and other technical societies on the matter, he should have been led to a conclusion which is contrary to what has hitherto been accepted, namely, that the actual improvement that it is possible to bring about on the water side of a boiler tube cannot sensibly affect the overall rate of heat transmission from the gases to the water. It will be well, therefore, in the first place, to consider the exact conditions under which some of the experiments quoted were carried out before accepting the quantitative conclusions derived from them. For example, in Box's experiment, the water is initially at rest, and due to the absorption of heat, molecular convection begins, then increases and reaches a maximum during ebullition. Hence, on the water side, the rate of heat transmission is practically zero to begin with, but increases to a maximum; the overall effect must necessarily be large. On the other hand, in an actual feed water heater, the water has an appreciable positive velocity at the start, and therefore a definite rate of heat transfer is available, which, at a speed of about 2¾ feet per second is approximately 0.5 B.T.U. per second per square foot per degree F.* On the steam side, the value of the rate of heat transmission is about 1, for condensing steam. It seems reasonable to assume that the maximum rate of heat transmission possible on the water side when generation of steam takes place will be about 1 also. From which it will appear that the overall rate of heat transfer will be increased from

$$\frac{1}{\frac{1}{1}+\frac{1}{.5}} = 0.333 \text{ to } \frac{1}{\frac{1}{1}+\frac{1}{1}} = 0.5,$$

that is, an increase of about 50 per cent. From Figure 1 an increase may be expected from 0.061 at 90 degrees F. to 0.186 B.T.U. per second per square foot per degree F. at 212 degrees F., or an increase of 200 per cent. First then, the overall gain which may be due to temperature effect is not always as high as Mr. Andrews believes, and secondly, comparison of the values 0.061 and 0.333 for the same water temperature indicates how essential it is to examine all the factors which may affect the results of one particular experiment before passing on to a generalization of these.

Again, in Figure 2 the slope of the curves represents heat lost per unit of time by the hot water, and presumably Mr. Andrews ascribes the flattening of the curves at the lower temperatures to a decrease in the rate of heat transfer. But the curves indicate overall changes, and consequently include the increasingly beneficial influence from the cold water side. The flattening is, in fact, simply due to the diminishing difference in temperature which exists between the hot and the cold sides.

Having satisfied himself from the "very crude experiment" of Box (which it will be remembered is of an overall character) that "as water becomes hotter . . . it absorbs heat more rapidly than it does in the cold state" (which is a component effect), Mr. Andrews proceeds to apply this deduction to a boiler and states his conclusion as the missing link in the theory of heat transfer. Considering the rapid evolution which has taken

* Webster. Trans. I. E. S., vol. LVII, p. 58.

place in boiler design since the days of Nicolson and other heat transmission experts, it is rather difficult to accept the explanation offered by Mr. Andrews, and to believe that designers of steam boilers have never reasoned on the lines laid down by him.

Mr. Andrews has completely overlooked the fact that the component transmission rate on the gas side is only about 0.005* B.T.U. per second per square foot per degree F., and on the water side this component may vary from about 0.5 to 1. Therefore, from calculations similar to those shown for the feed heater, the overall effect from the gas to the water side can only increase from 0.00495 to 0.00498, that is, about 0.6 per cent. As the component water rate 1.0 corresponds to the condition of actual generation of steam, it follows that when a certain amount of heat is to be abstracted from a gas in a given time, it makes little difference to the cooling surface whether this heat is used to heat water or to raise steam. Mr. Andrews' missing link becomes now a mere phantom which may vanish altogether when it is considered that temperature difference as well as rate of heat transfer enter into the final determination of heating and cooling surfaces.

A boiler may be regarded as made up of two portions, one responsible for the heating of the feed water, an economizer in fact, and the other responsible for the generation of steam. The latter is evidently unaffected by what may happen in the former. The first part of the boiler is, therefore, alone susceptible to being reduced, and consequently it is very important to know exactly how the feed is to be preheated before concluding that there will be a definite reduction of surface. Taking the three alternatives given on page 649, it will be agreed that there will be a definite gain by the utilization of exhaust steam, but when boiler steam is to be used, there is no appreciable gain since the surface which acted as economizer is now required to produce the steam necessary to heat the feed water.

The alternative of using bled steam is also open to discussion, because it involves consideration of the turbine side of the steam cycle. It will be sufficient to indicate that here again the output of the boiler must be increased for a given horsepower from the turbines.

There are many other points in this paper which require explanation, but as I have already dealt at some length with a few, I shall conclude by asking those who are using desuperheated steam for auxiliaries to remember that superheated steam is a gas, with a low rate of heat transmission,† and therefore desuperheating steam which has been produced at the expense of relatively large and costly heating surfaces, seems an unnecessary waste which can be avoided by extracting saturated steam from the boiler drum.

Mr. G. E. Carter: The paper is particularly valuable at the present time when the Diesel engine has galvanized the steam engine into fresh activity in order to compete with the Diesel in the matter of economy. In the case of reciprocating steam engines having a low-pressure turbine incorporated, the turbine can make use of steam at a temperature at which it is of little use for raising the feed to any appreciable heat. On ships of this type live steam feed-heating is necessary, especially if, as in some cases, funnel temperatures are kept low by heating the air supply to boilers by the funnel gases.

As feed heaters of this type work on a closed system, necessarily well above atmospheric pressure, some remarks by Mr. Andrews on methods for preventing air, entrained in the feed water, entering the boilers, would add to the value of the paper, as in some cases surface contact heaters have been blamed for causing corrosion in the boilers.

Mr. George E. Tansley, B.Sc.: There is one point which appears to have been overlooked by Mr. Andrews, and that is that no matter how efficient

* "Preheated Air for Combustion." Journ. R. T. C. 1926.
† Actual figures will shortly be published in Journ. R.T.C. 1930.

a boiler or economizer may be on the water or wet side of the plate, the rate of heat transmission is governed entirely by the rate at which the heat can be transmitted from the gases to the plate. It is well known that with water on one side of a plate or tube and gases on the other side, the water will absorb many times the amount of heat which could possibly be transmitted to the plate or tube by the gases. This is clearly demonstrated by Mr. Andrews himself when he gives the amount of heat which can actually be absorbed by the water when the obverse side of the heating surface is in contact with the steam.

It may be true that hotter water will be in a state of greater turbulence, and that the capacity for sweeping off more heat from the wet surface will be increased, but unless the heat can be in some way transmitted from the gases to the plate in such quantities that this added efficiency on the water side can be taken advantage of, this additional capacity for extracting heat from the wet side of the plate is of practically no value. It is easy to believe and visualize the added advantage when dealing with feed water heaters in which the heating medium is either exhaust or live steam, but that is a totally different problem. It seemed it as well to mention this fact as it might eliminate any possible confusion of thought when discussing the later figures in this very interesting paper.

Mr. R. Royds, M.Sc.: To my mind the remarkable feature about this paper is that it contains or revives many fallacies which I thought were long ago dead and buried. A few examples taken from the paper will no doubt suffice to explain this contention. On page 647 Mr. Andrews apparently suggests that were it not for a coating of soot the transmission of heat from gases to water would be about six times as great with copper heating surfaces as with steel surfaces, and he suggests that what is suitable for feed heaters in this respect ought to be equally suitable for boilers as far as the transmission of heat is concerned. It was thought that this fallacy had been already sufficiently exposed, but apparently not so. To restate the position, there is no doubt but that the heat transmitted from gas to water with copper plates or tubes cannot be appreciably different from that with steel, because the resistance of the metal to heat flow is only a very small proportion of the total resistance from gas to water. With feed heaters, however, the resistance of the metal may be an appreciable proportion of the total resistance between the steam and water. For example, if the total resistance from steam to water is represented by 40, while that of the steel tube is represented by 6 and the copper tube by 1, using copper instead of steel would reduce the total resistance by 5 in 40, or, say, about 13 per cent. On the same scale the total resistance from gas to water may be about 1000, and using copper instead of steel would only reduce the total resistance by about 5 in 1000, or about ½ per cent. In the boiler problem the major part of the resistance to heat flow is that between the gas and the surface, and any appreciable reduction of the total resistance must be obtained on the gas side.

Again, take the supposed results of Mr. Andrews' experiment exhibited in Figure 4. He mentions a gas temperature of 900 degrees F. When the water is at 212 degrees F. the temperature of the metal will be about 220 degrees F., and the heat transmitted per unit of surface per unit of time works out at $(900 - 212) \times 13.3 = 9150$, according to his results, and per degree difference gas to surface would be about $\frac{9,150}{(900 - 220)} = 13.5$. With feed water at 60 degrees F. the corresponding heat transmission is, according to Figure 4, $(900 - 60) \times 6 = 5050$. Nothing has been altered on the gas

side either with respect to temperature or rate of gas flow, and thus the rate of heat transmission on the gas side is as before, 13.5. Thus the temperature of the metal, T_m, is now given by

$$(900 - T_m) \times 13.5 = 5050$$

or, $T_m = 526$ degrees F.,

a result which is most absurd, and without doubt shows that Mr. Andrews' experiments on this matter are quite unreliable. In fact, this gives an ample illustration of the dangers attending rough and ready experiments made without taking adequate precautions to ensure a reasonable degree of accuracy.

Further, Mr. Andrews quotes some boiler experiments. For instance, a boiler evaporating 12,000 pounds of steam per hour without feed heating becomes 26,080 pounds per hour after feed heating to 384.5 degrees F. was introduced. But the velocity or weight flow of the gases cannot be the same in both cases, and Mr. Andrews does not seem to know, or has suppressed the knowledge, that the influence which gas flow has on the evaporation is most important.

Live steam feed-heating may be obtained by suitably exposing the feed water to the steam in the boiler drum before allowing it to mix with the body of water, and hardly needs a separate surface feed heater for the purpose.

Mr. A. W. JOHNSTON: Mr. Andrews has done well in presenting his interesting paper on the subject of feed water heating, although his conclusions may not all be accepted. Nos. 1 and 3 are unquestioned. No. 2 is a probability with certain limitations. No. 4 is a matter involving other factors in addition to feed water temperature. No. 5 appears to have been proved in practice, and No. 6 is probably more incidental than real, depending possibly upon higher feed temperature affecting other functions of the steam-generating equipment. If live steam heated feed water had intrinsic value, it would appear that the feed heating function should be performed in a part of the boiler steam space specially set apart for such purpose, thus cutting out the inevitable heat losses in the pipe work and external feed water heater. With high-pressure boiler plant the steam space is already restricted due to constructional difficulties, and an external heater may be justified in such cases.

It would be interesting to have tests made on boilers and superheaters combined, before fully accepting the claim made of 62.7 per cent increase in generating capacity due to feed water heating, as in the case of the Wood Lane boiler. A test was made some time ago on a boiler operating at full load with a feed temperature of 200 degrees F. The boiler (oil-fired) was then tested with feed water at 250, 300, and 350 degrees F. by passing the feed through a live steam heater. The furnace temperature and flue-gas exit temperatures and also the superheat temperatures were observed. The oil-fuel rate at the burner and the draught were constant during these tests. It was found that as the feed temperature was increased a slight drop occurred in the maximum furnace temperature, the exit flue-gas varied only a few degrees, while the superheat temperatures dropped at an increasing rate at the higher feed-water temperature, to such an extent at 350 degrees F. feed temperature that it was deemed inadvisable to raise the temperature of the feed water to 400 degrees F. for which provision was made.

These results appear to indicate that increased evaporation occurred on the radiant heat absorbing surfaces, and the loss in superheat was due to decreased gas temperature in the superheater zone simultaneously with the increased quantity of steam passing through the superheater, due to the

higher feed temperature. The flue-gas exit temperatures indicated that no thermal saving could be ascribed to the higher temperature of the feed water.

Mr. J. RUSSELL LANG (Member): At a time when all steam engineers are looking for economies in the production of power for either marine or land work, Mr. Andrews' paper, giving us a new aspect of feed water heating, is of great interest.

It is well known that from a purely thermodynamic standpoint, there is no gain in heating feed water with live steam, but there is a definite economy in practice. This gain results from better circulation in the boiler and from the fact that more heat is absorbed by the same heating surface, and a lower flue-gas temperature results. In a modern power station, the maximum economy is obtained with three- or four-stage feed heating, useful work being done by the steam before it is used in the feed heater. The fourth-stage feed heater may use saturated steam at a pressure of 200 pounds per square inch bled from the main turbine, so that very high feed temperatures are possible without using live steam for feed heating. Further, when the feed temperature is further increased by an economizer, it will be seen that the final feed temperature is not so far short of the boiler temperature as Mr. Andrews indicates. The curves shown in Figures 1 and 2 simply indicate the well-known phenomenon of the effects of viscosity and turbulence on heat transmission; no mention is made of the sudden jump in transmission rate when the flow through a tube suddenly becomes turbulent on increasing velocity.

In his comparison between copper and steel Mr. Andrews omits the fact that the heat flow from one fluid to another across a metal membrane does not depend entirely on the relative conductivity of the metal. The flow is more dependent on the resistances offered at the surfaces, that is, the resistance offered to the passage of heat from the one fluid into the metal surface and from the metal surface into the other fluid. The actual resistance through the metal is only one factor in the total resistance. These added together amount relatively to about 6 for steel and 10 for copper for a normal surface feed water heater. These figures are very different from those given for the actual conductivity of the metals themselves. A similar problem is encountered in dealing with evaporators where heat is transmitted from steam inside the coils to water outside. Heater tubes of copper have been shown which give from 15 to 20 per cent better heat transmission than brass tubes, that is, copper is practically 20 per cent more effective than brass, and 20 per cent less surface is required for the same work.

It is regrettable that Mr. Andrews chose such low steam pressures as are given in Table I. No installations of any size or note are now built with such pressures. Mention is made of modern large passenger vessels requiring enormous boiler installations. This is not the case, as these boiler installations are, in fact, very small for their power; they are of the water-tube type, having large combustion chambers, fired by oil fuel, and as previously noted have the feed heat carried to the highest economical limits.

Reference has been made in the discussion to a recent vessel fitted with a live steam heater in addition to two other stages of feed heaters. It is presumed that the vessel referred to is the *Darien*, and if so, it should be noted that the live steam feed-heater receives steam from a donkey boiler and condenses it to form the feed make-up. The feed is raised from 290 to 310 degrees F. in this heater, steam being supplied at a pressure of 140 pounds per square inch.

Mr. W. H. WOOLNOUGH: Mr. Andrews has put before us in a very simple statement a collection of leading principles for use in the consideration of steam-raising plant. These he has conned from his experience in engineering, and from his knowledge and appreciation of other great workers of his earlier days, of whom, unfortunately, much too little is heard today.

Mr. Andrews has dealt with what is, to many, a new aspect of feed heating, but reference might be made to the great service he has done in respect of the excellent qualities of the fed water heaters he has produced. According to my observation, each of his designs has, in its day, been unsurpassed, and he has usually been years ahead of other designers in his perception of the solution of our feed heating problems. Mr. Andrews' deductions from the tests of the Druitt-Halpin system are to me exceedingly interesting, because I have made use of the same general ideas in boiler design, and with very satisfactory results.

On page 648, Mr. Andrews gives the rates of heat transmission in a typical boiler and in a live steam feed-heater. The rate quoted for the boiler is, as he evidently intends to indicate, an average rate for the whole boiler and would fit the case of some recent three-drum marine boilers. That the maximum rate is often much greater than his average is evident from his reference to Bryan & Donkin's work. For example, assuming an average rate of 4600 B.T.U., the rate through 10 per cent of the surface would be about 30,000 B.T.U. It may be of interest to note that in some recent boilers, various considerations compelled me to design for an average heat-transmission rate of 23,000 B.T.U. per square foot in service, and on trial an average rate of 34,000 B.T.U. was attained. Thus the maximum rate must have approached Mr. Andrews' figure for feed heaters. While it is only fair to state that such figures would be unattainable were not full use made of the radiant heat of the fire, the average rate appears to indicate results far in excess of Mr. Andrews' basic figure.

Mr. Andrews' conclusions are very well worth remembering, but if any one may be selected for special note, item 4 seems to me very important, for if regarded comprehensively it does lead to major improvements in the design of steam-raising plant. A boiler having an efficiency of say 85 per cent may be regarded as a good boiler, but a boiler combined with feed heaters, having the same overall efficiency can have advantages in cost and weight and space occupied, rendering it much superior.

Mr. C. WALDIE CAIRNS, M.Sc.: To me the most important point insisted on by Mr. Andrews is that the higher steps in feed heating make a great difference in performance, in the sense that they increase the efficiency of the heating surface, or of a part of it in extracting heat from the gases.

Box's experiments quoted in the paper, and those made by Mr. Andrews himself, are interesting, even although made in open vessels and under conditions which complicated the results by the combined effects of convection currents, of local temperatures of the water, and of ebullition or of incipient ebullition. Nevertheless it may no doubt be safely assumed that water, when actively boiling, will take heat best out of metal—provided the steam bubbles have room to get off and sweep water along with them, or when on the point of boiling, in which condition steam bubbles are no doubt being formed, and being condensed in adjacent water induce a lively motion in water which is already at its minimum viscosity.

But it is very difficult to estimate to what extent very high feed temperature, however produced, would allow of a reduction in boiler heating surfaces without loss of boiler efficiency, as measured by fuel consumption. In water-tube boilers, and in locomotive boilers, in which the water content is comparatively small in relation to the evaporation per hour, the average temperature throughout the water in the boiler may be kept quite notably below the temperature of evaporation by cold feed-water, especially in "prehistoric" plant such as Mr. Andrews cites, where water-tube boilers are ill-treated with feed water at 100 degrees F. In such cases the substitution of very hot feed from a thermal storage tank or from a live steam feed-heater

(especially if this heater follows an added waste-heat heater) might well make a difference. In the case mentioned of thermal storage as an aid to peak loads, however, we are asked to accept a most remarkable increase of capacity for load as being due to feed temperature alone, without any aid from waste heat. Really to understand these examples we ought to have the boiler efficiencies at the two loads, or at least, the flue-gas temperatures at the exit from the boilers at the two loads. These data are not given in "The Engineer" article quoted by Mr. Andrews. At the peak load, these boilers must have been burning much more fuel than at normal load. There would consequently be higher gas temperatures all through the boiler, accounting for much of the increased output; and unless the effect of the hot feed was almost miraculous, there must have been a higher temperature of gases at the exit from the boiler, resulting in a reduction of boiler efficiency in terms of water and fuel.

The extent to which heating surface can, with advantage, be provided in a live steam feed-heater, instead of in the boiler must be determined by the limit below which a further reduction of boiler heating surface under ordinary working conditions would produce a rise of temperature of flue gases leaving the boiler. As Mr. Andrews points out, the amount of heat to be transmitted by the heating surfaces of the boiler is exactly the same whether a live steam heater is used or not, leaving out of account any additional radiation losses involved in steam and drain piping and in the heater, though these need not be great with good lagging. In the absence, therefore, of complete and full-scale experiment, cautious procedure is necessary.

It is claimed in the case of the Wood Lane station boilers that although they primed badly at 15,000 pounds per hour with cold feed, they produced 26,080 pounds per hour without priming when on hot feed. This is difficult to account for, unless the cold feed contained impurities liable to produce foaming. Such impurities might be deposited, among others, in the Druitt-Halpin thermal storage tanks. The tanks, too, would contain a very appreciable proportion of condensed steam of high purity.

Mr. Andrews revives the old comment on the small amount of heat transmission achieved at certain parts of the heating surfaces of locomotive and cylindrical boilers. It must, however, be kept in mind that some part of the surface must deal with low-temperature gases, and with non-radiant heat, and, as Mr. Andrews quoted from the Blechynden experiments, rate of heat transmission is lowered by every decrease of gas temperature. Mr. Andrews is no doubt right in claiming that hot water on the other side will help these low-temperature surfaces to do their best, but it seems to me that in the modern cylindrical boiler, into which the feed water, at about 200 degrees F., is introduced by internal feed pipes just above the tube nests, this feed, mingling as it does with a lively current carrying a great volume of hot water, must attain a very high temperature before it reaches any heating surface. In a normally worked cylindrical boiler from about 8000 to 10,000 pounds of feed water per hour, say, 14 to 17 gallons per minute, of feed are pumped into a boiler containing from 20 to 25 tons of hot water in very lively circulation. I should, therefore, be very much surprised at any substantial gain of efficiency or of steaming capacity from an increase of feed temperature from a normal 200 degrees F. to, say, 10 or 20 degrees below the evaporating temperature in the boiler, such as might be attained by an external live steam feed-heater. No doubt a rise up to practically evaporating temperature could be attained by using a suitable contact heater inside the steam space of the boiler, and this would give the last few degrees rise which Mr. Andrews suggests is so valuable.

Table I, giving marine boiler results, is interesting, but these were cases where previously most primitive arrangements had apparently been in use,

and it must be realized how difficult it is to get or carry out valid comparisons even between successive voyages of the same ship. Very few people seem to realize that, other things being equal (and how seldom they are equal), one revolution off or on 70 R.P.M. should account for over 4 per cent of the fuel consumption, and that a change of direction or force of quite moderate degree in the wind can account for four or five times that percentage.

It can, I think, be agreed that the hotter feed water is put into boilers, the better for steaming and durability, but information will yet have to be accumulated as to the extent to which money can be diverted from the provision of heating surface in boilers to heating surface in live steam feed-heaters, so that economy may be maintained both in capital cost and in fuel cost.

Meanwhile some small-scale experiments on heat transmission at various temperatures might well form the subject of some post-graduate research. The plant need not be costly, provided means are already available for the measurement of a flow of water, for the basis of the method would have to be the measurement of a small rise of temperature in a large flow of water, so that temperature range of the heated water would not be great, and the condition constant for sufficient time to justify definite figures. Such experiments might be carried out under pressure, and up to temperatures approaching or equalling boiling point for each pressure dealt with. The conditions could be varied so that convection effects were either included or excluded, and by the use of a heating element consisting of a large-bore tube fed with saturated steam and kept absolutely free of condensate accumulation, the heat supply could be kept at any desired steady temperature, so that the temperature effect would only vary on the heated water side. Judging from reference books I have consulted, exact data of the kind which would be yielded by such experiments appear to be necessary but not available, and it is to be hoped that the importance of the subject, as demonstrated by Mr. Andrews' paper, will lead to some exact experiments. Apart from feed heating and boiler questions, there is a large and increasingly important field of industry depending largely on the heating and evaporation of liquids, in which such data would probably be useful.

AUTHOR'S REPLY.

Mr. ANDREWS: It is exceedingly interesting to learn of Prof. Cormack's experience of the remarkable results obtained with the Druitt-Halpin thermal storage system. Such testimony is of considerable value to the paper coming as it does some 38 years after the event. It is also desirable that Prof. Cormack's recommendation to undertake experiments on a full-sized boiler should be followed, so that marine steam machinery may obtain the full benefit of higher feed temperatures.

Mr. Fulton seems to agree generally with the substance of the paper, and comments upon the extent to which feed heating is now being applied. He mentions 310 degrees F., as having been recently applied, but the latest proposition which has come under the author's notice is to raise the feed temperature to 450 degrees F. before putting it into the boiler. Even then, however, it is considerably under the steam temperature on account of the high pressure involved. He asks why the boiler is not reduced in size by 30 per cent if it is possible to do so. The answer is that the fact was unknown until this paper was written. Then, he raises two objections that are more fictitious than real. There is nothing more elaborate or costly about the substitution of feed heating surface for boiler heating surface, because, at most, one square foot of heater surface transmits as much heat as ten square feet of boiler surface. Moreover, the heater surface is much more

compact than the boiler surface, and is much more easily stowed in the machinery space of a ship. Further, the temperature around a feed water heater is not nearly so high as that around a boiler, and therefore there need be no fear of inability to insulate it easily.

The Wood Lane boilers were not highly forced when providing 26,080 pounds of steam per hour from hot feed water; they were said to be working in a manner somewhat similar to that of their rated production with cold feed water. It would not be easy to find experiments on the transmission of heat from steam to water and a dry gas to water under precisely the same conditions, because the dry gas is always at a much higher temperature than the steam, but, even with that advantage, there is ample evidence in favor of steam transmitting the greater amount of heat.

Dr. Sneeden's criticism seems to be based on what he says has hitherto been an accepted principle, namely, "that the actual improvement that it is possible to bring about on the water side of a boiler tube cannot sensibly affect the overall rate of heat transmission from the gases to the water." If, by "improvement on the water side" is meant an increase in the water temperature, then this accepted principle has been demonstrated in actual practice to be wrong since the introduction of the Druitt-Halpin system 38 years ago. When the feed water temperature at the Wood Lane station was increased from 100 to 384.5 degrees F., the evaporation was increased from 12,000 to 26,080 pounds per hour. If this increase of feed temperature had not increased the rate of heat transmission from the gases to the water, the increased evaporation would only have been 16,030 instead of 26,080 pounds per hour, which it actually was. The only observed change was in the temperature of the feed water, and the result proves beyond a doubt that the increase of the feed temperature did in fact increase the rate of heat transmission.

Dr. Sneeden's criticism of Figure 1 may be quite true, but it does not affect the argument attached to it in the paper, namely, that "the rate of heat transmission increased with each rise of the initial temperature of the water to be heated." The precise amount of this increased heat transmission does not affect the purpose. Dr. Sneeden is quite wrong when he states that the missing link is deduced from Firgue 1. Figure 1 is merely a part of the argument leading up to the actual economy obtained on marine boilers by the use of live steam feed-water heaters; and the missing link is the increased rate of heat transmission in the boiler due to the higher feed temperature. It is the only reason that has ever been adduced to account for the economy of live steam feed-heating.

The results of the experiments illustrated in Figure 2 were primarily given to show the influence of diffusion of the water on the rate of heat transmission, and Dr. Sneedon does not dispute that it was so, but he claims that the flattening at the lower end of the curves is simply due to the diminishing difference of temperature. While that has certainly some influence it is not the whole case, because in Figures 1 and 4 the rate of heat transmission rises rapidly notwithstanding the decrease of the difference of temperature. It may be difficult for Dr. Sneeden to believe that boiler designers have never reasoned on the lines laid down in the paper, but this discussion should convince him of the contrary. Boilers are still being designed with as much heating surface in relation to the water evaporated as ever they were, irrespective of the extent of feed water heating, and no one, so far as the author is aware, has ever attempted to apply the advantages of the Druitt-Halpin system to marine boilers, whereby the boiler heating surface may be reduced from 25 to 44 per cent without forcing the boilers in any way.

Mr. Carter, as a practical engineer, finds the paper to be particularly valuable in so far as it will help the steam engine to compete with the Diesel

engine. Certainly if full advantage be taken of feed water heating it should considerably reduce the first cost and running cost of steam machinery, both of which are most desirable in a country where coal is at hand and oil has to be imported. The method of extracting air from the feed water in connection with a closed surface feed-heater is the simplest and probably the most efficient of any in operation today. An air collector, fitted with a gage glass, is connected to the heater outlet. As the water passes this collector it gives off its air in precisely the same way as the air from a column of water fills its air vessel; but the air, being hot, is at its maximum volume and separates from the water more freely than if it were cold. The actual collection of the air is visible in the gage glass, and can be adjusted to blow into the atmosphere.

Mr. Tansley claims "that no matter how efficient a boiler or economizer may be on the water or wet side of the plate, the rate of heat transmission is governed entirely by the rate at which the heat can be transmitted from the gases to the plate." That is but a slight variation of Dr. Sneeden's way of putting it, and the answer to it is the same and need not be repeated. It may be mentioned, however, that the amount of heat which can be transmitted per unit of surface in a feed heater is considerably in excess of that given in the paper. The figure given is a very moderate one to avoid any appearance of exaggeration.

If Mr. Royds has got the impression from page 647 that soot is the only impediment to the transmission of heat, or that a copper heating surface in a boiler should transmit six times as much heat as a steel surface, that impression was not intended to be conveyed. Nor is it suggested that what is suitable for feed heaters ought to be equally suitable for boilers as far as the transmission of heat is concerned. At the same time, the author cannot agree that the rate of heat transmission cannot be appreciably different between copper and steel tubes in a boiler. When water-tube boilers were originally fitted in torpedo-boat destroyers 40 years ago, it was specified that the boilers should have three rows of copper tubes on each side next the fire, the remainder being of steel. Of the three vessels, for which the author designed the boilers, not one of them reached the trial trip with copper tubes, because they burst repeatedly before full power was reached, and eventually, as the result of a serious fatal accident, all the copper tubes were removed and steel tubes substituted. There was no question about the quality of the copper, which was tested by Admiralty officers; the fault was wholly attributed to increased heat transmission and excessive circulation, but as the circulation was the same for copper and steel tubes, Mr. Royds' ½ per cent difference is obviously at variance with practical experience.

Mr. Royds repeats the proposition which has already been made twice during this discussion with slight variations, namely, "that the major part of the resistance to heat flow is that between the gas and the surface, and any appreciable reduction of the total resistance must be obtained on the gas side." The discrepancy between this statement and the actual results obtained by the Druitt-Halpin system is not merely appreciable, it is enormous, and the economy obtained by various processes of feed heating, which is rapidly extending at the moment, is also more than merely appreciable. It is a curious circumstance that those who have divided the process of heat transmission into component parts, have criticized the paper from the one point of view, and have not attempted to show that any other component in the process would account for an increased evaporation of 117 per cent. Mr. Royds' concluding statement about internal boiler-feed heating has been many times conclusively proved to be wrong, but he need only compare the 8 per cent gain in Prof. Gibson's careful experiment with the enormously greater gain in the Druitt-Halpin system.

Mr. Johnston's appreciation of the paper is encouraging to future progress. It is hardly to be expected that he would accept all the conclusions resulting from the paper, but it is satisfactory to know that he accepts most of them; and, without knowing the precise reason for his limitation of the others, it would not be wise to speculate. As regards conclusion 4, however, there can be no doubt that if hot water is put into the boiler and the boiler heating surface has that much less heat to transmit, then the boiler heating surface can be reduced while retaining the same rate of heat transmission per square foot of surface. It has been repeatedly shown in the cylindrical boilers of British warships, and particularly by Prof. Gibson's test already referred to, that live steam feed-heating in the steam space of the boiler can only produce a relatively small economy, because the element of time to heat the water is lacking and its influence on the dryness of the steam has to be considered. Certainly the test suggested by Mr. Johnston would be interesing if carried out with care on a single boiler; but, as far as this paper is concerned, the addition of superheating the steam would complicate the problem and perhaps obscure the true effect of both feed heating and superheating.

Mr. Lang's opening remarks are generally in agreement with the author's, and the slight difference of view has already been dealt with in reply to other speakers. The steam pressures dealt with in the paper are those in common use in modern cargo and passenger vessels; moreover, they were similar to the pressure used in the Druitt-Halpin system, and, in any case, the pressure does not appear to affect the principle. Regarding the difference in the rate of heat transmission between brass and copper tubes in feed water heaters, Mr. Lang's figures of 15 to 20 per cent in favor of copper may be quite correct when tried on a given heater, but much depends upon the design of the heater. In the author's experience the difference in favor of copper is considerably greater with a good design of heater and fairly clean tubes. Mr. Lang asserts that modern boiler installations are very small for the power. That may be in so far as modern machinery is more economical in steam consumption, but it is not the case as regards the rate of evaporation per square foot of heating surface, as last session's Transactions of the Institution will show.

Mr. Woolnough has provided a respite among the critics, for which the author is thankful. Mr. Woolnough has had considerable experience of feed water heaters, and it is very encouraging to get this acknowledgment of his satisfaction. It is also pleasant to find that, having made use of the same general ideas in boiler design as those relating to feed heating in the paper, he has had very satisfactory results. This practical testimony is more valuable than any amount of speculation. He is quite correct in assuming that the rate of heat transmission given for a typical boiler is an average rate to be commonly found on cargo and passenger vessels. There are special types of boilers, like the first Wood steam-generator for example, and the type to which Mr. Woolnough refers in which very high rates of heat transmission have been obtained by special means, but those special cases are of limited application at the present time and could not be dealt with in this paper. It is very satisfactory to have Mr. Woolnough's confirmation of the importance of conclusion 4 because it has never been put into practice, and there must be a considerable saving in boiler heating surface by applying the principle.

Mr. Cairns puts the case of heat transfer to the water in excellent terms. His difficulty about reducing the boiler heating surface has already been dealt with in connection with conclusion 4. The boilers on which the Druitt-Halpin tests were made were neither "prehistoric" nor ill-treated boilers. They were thoroughly good water-tube boilers of a type which is made to the

same general design at the present time; and Mr. Cairns need have no compunction about accepting the figures for the increased evaporation, because, although "The Engineer" did not publish all the information that would satisfy everybody, still it was made perfectly clear that the conditions were generally the same, except in the temperature of the feed water. But a most fortunate circumstance for the author is that Prof. Cormack had also tested the Wood Lane boiler and is able to confirm the results obtained by Druitt-Halpin and Prof Unwin; and also to confirm the author's statement that "there can be no doubt about the facts." If Mr. Cairns will accept such unimpeachable testimony most of his other doubts must go by the board, because this plant was on land. It was not affected by the vicissitudes of a sea voyage or the imperfect records of a second or third engineer. It is agreed that nothing but a trial on a full-sized boiler using increasing feed temperatures will demonstrate the true facts.

The author has to thank those who took part in the discussion; and if this contribution to the Transactions should stimulate interest in the subject to the benefit of steam machinery, the author will be amply rewarded. —"Institution of Engineers and Shipbuilders in Scotland," Vol. LXXIV, 1930-1931.

PROPELLING PLANT OF THE *KÖNIGSBERG*-CLASS OF GERMAN CRUISERS*.

TRANSLATED BY E. C. MAGDEBURGER.

The principal data with regard to the three small cruisers *Koenigsberg, Karlsruhe* and *Köln,* constructed by Germany in accordance with the Versailles treaty are as follows:

Displacement (without fuel and water supplies)			6,000 tons
Length between perpendiculars—L	169.0	meters	554.47 feet
Overall length	174.2	meters	571.53 feet
Maximum breadth	15.3	meters	50.20 feet
Draft (design)	5.3	meters	17.39 feet
Freeboard—H	9.25	meters	30.35 feet
L ÷ H	18.3		18.3
Main turbine propelling plant			65,000 S.H.P.
Cruising Diesel propelling plant			1,600 S.H.P.
Designed maximum speed			32 knots

Battery—9-15 centimeter (5.9 inches) guns in triple turrets.
4-8.8 centimeter (3.45 inches), 8-3.7 centimeter (1.46 inches) guns and four machine guns.

Torpedo equipment—12-50 centimeter (19.7 inches) torpedo tubes in triple tube units.

The main propelling plant of these cruisers designed for a steam pressure of 242 pounds per square inch, consists of six double ended oil fired boilers (two of 13,500 square feet and 4 of 10,900 square feet heating surface) and four independent two-cylinder steam turbine units driving the two propeller shafts through reduction gears. One pair of units constitutes the main turbines, the other pair the cruising turbines and they are so arranged that either of the units or both together can drive the shaft. For further details of machinery arrangement see Figure 1.

To this geared-turbine propelling plant has been later added an economical oil engine cruising plant, which consists of two very lightly built ten-cylinder Diesel engines, developing about 800 S.H.P. at about 900 R.P.M.

* Excerpts from a descriptive article by W. Laudahn in Schiffbau of 1 Feb. and 15 April 1931, pages 49 and 181.

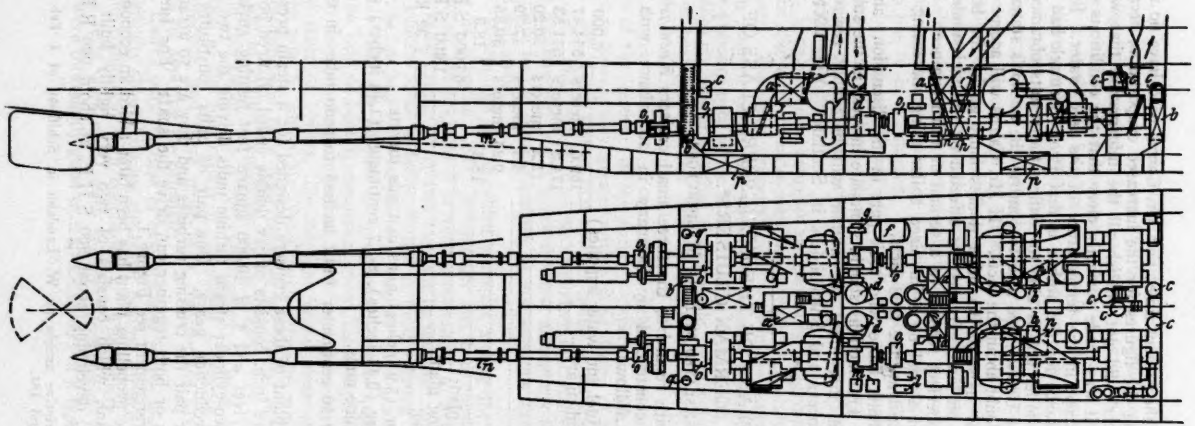


FIG. 1. MACHINERY ARRANGEMENT OF CRUISER "KÖNIGSBERG."

a Hot well
b Lub. oil pumps
c Lub. oil coolers
d Evaporator, incl. pumps and coolers
e Fresh water condenser
f Aux. condenser
g Aux. circ. water pump
h Aux. air pump
i Steam sanitary pump
k Motor driven fire and bilge pump
l Mot. dr. gravity pump
m Steam torpedo air compressor
n Torsionmeter
o Disconnecting jaw coupling
p Lub. oil sump tank
q Gravity tank for Vulcan gear of oil engine drive

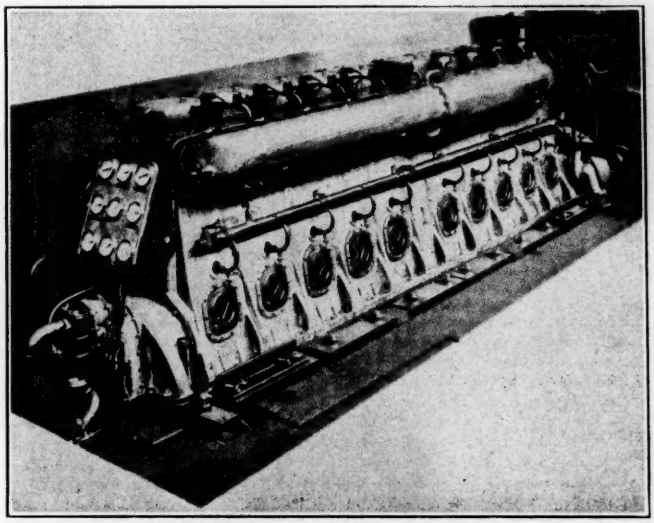


FIG. 2. CRUISING DIESEL ENGINE ON THE "KÖNIGSBERG" CLASS OF CRUISERS, 800 B.H.P. AT 900 R.P.M. NORMAL AND 1000 B.H.P. AT 1000 R. P. M. MAXIMUM RATING.

and a maximum of about 1000 S.H.P. at about 1000 R.P.M. This cruising plant was not provided in the original design of these cruisers and the Diesel engine was developed for a different purpose. But when this type of engine, which weighs only 5.5 kilograms (12.15 pounds) per brake horsepower, gave such brilliant results on test stand of the builder and particularly demonstrated a high degree of reliability, it was deemed highly desirable to use these engines in the new cruisers to give them thereby a highly economical propelling plant for cruising—even if only at such a slow speed

	Königsberg	*Karlsruhe*	*Köln*
Building Yard	Navy Yard Wilhelmshaven	Deutsche Werke Kiel A.G.	Navy Yard Wilhelmshaven
Construction began	12-4-1926	27-7-1926	7-8-1926
Date of Launching	26-3-1927	20-8-1927	23-5-1928
Builder of Propelling Turbines	Schichau, Elbing	Fr. Krupp, Germania Werft, A. G. Kiel	Blohm & Voss Hamburg
Builder of Cruising Diesel Engines	M. A. N. of Augsburg		
Date of Commissioning	17-4-1929	6-11-1929	15-1-1930

as 10—10.5 knots—and thus increase immensely their cruising radius. Some space, although not very generously dimensioned, could be made available in the after part of the vessel and the very light weight of the units permitted their use without exceeding the limits prescribed by the treaty of Versailles. It was, therefore, decided to incorporate these Diesel units even though they would not contribute anything to the attainment of the full speed of the vessel and would have to be carried as dead weight under these conditions. In the later design of the cruiser *Leipzig* now building even this drawback has been corrected, as it is hoped to show at some later date when the vessel is completed.

The cruising Diesel engines are of the reversing, single-acting, four-cycle type with mechanical injection of fuel, having working cylinders of 260 millimeter (10.236 inches) bore and 330 millimeter (12.992 inches) stroke. They work unsupercharged and transmit their output through a hydraulic clutch and reduction gear (known as a Vulcan gear) to the propeller shafts. The hydraulic *Vulcan* clutch prevents impacts and torsional oscillations from being transmitted to the rather sensitive in this respect gears. The gear ratio is about 7.65 to 1. When the steam turbines are operating the cruising Diesel engines are disconnected by means of the hydraulic clutches, while if it is desired to operate on oil engines these must be connected and the turbines disconnected while the shafts are at a standstill.

Figure 2 shows a three-quarter outside view of the cruising Diesel engine on the test stand. The location of the cruising oil engines is shown schematically on Figure 1 while Figure 3 shows the detailed arrangement of the Diesel engine room together with all the auxiliaries required by this installation. All pumps, such as for lubricating oil, fuel oil and circulating water, are direct driven by the engine itself and a geared lubricating oil pump serving the *Vulcan* gear is attached to the primary or engine driven

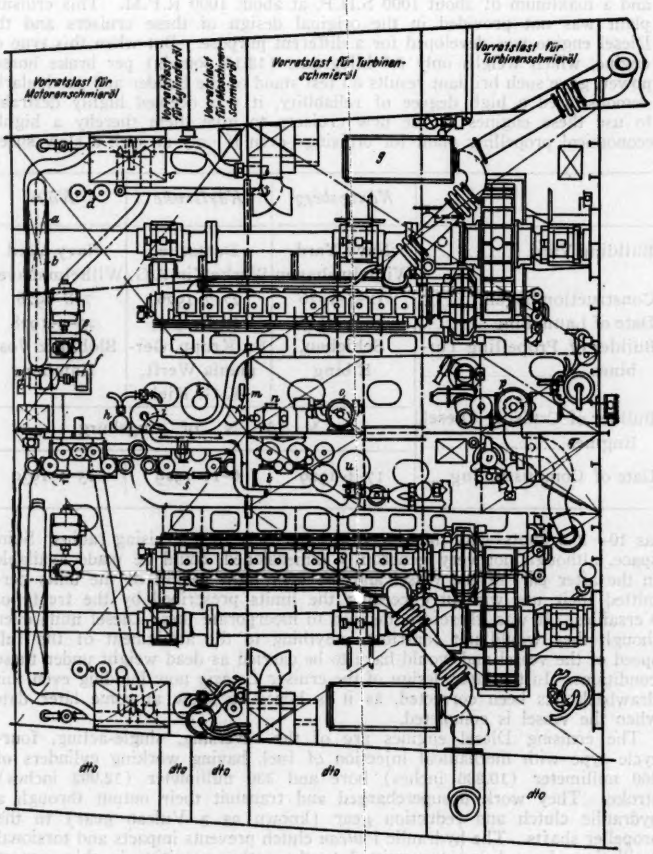


FIG. 3. ARRANGEMENT OF MACHINERY IN DIESEL ENGINE ROOM OF "KÖNIGSBERG."

a Fuel oil tank	h Lub. oil strainer	p Turbine oil sump
b Starting air flask	i Lub. oil cooler	q Drain pump
c Fuel oil day tank	k Master compass	r Change-over manifold
d Fuel oil strainer	l Cooler	s Change-over manifold
e Auxiliary pumps	m Manifold	t Bunker oil transfer pump
f Fuel oil transfer pump	n Transformer	u Air chamber
g Muffler	o Aux. lub. oil pump	v Aux. air compressor

Vorratslast für Motorenschmieröl — supply of Diesel engine lubricating oil.
Vorratslast für Zylinderöl — supply of cylinder oil.
Vorratslast für Maschinenschmieröl — supply of machine oil.
Vorratslast für Turbinenschmieröl — supply of turbine lubricating oil.

part of the hydraulic clutch. An auxiliary lubricating and fuel oil pump unit, motor driven, is located in the port engine room aft and each of the engine rooms has its own motor driven starting air compressor. Besides, in each of the cruising engine rooms there are the following auxiliaries installed:

One fuel oil transfer pump in starboard engine room only.
One drain pump in port engine room only.
One fuel oil tank.
One 200 liter (7.063 cubic feet) starting air flask.
One oil cooler for the oil of the hydraulic coupling.
One lubricating oil cooler for the engine oil.
One muffler.
One twin strainer for fuel oil.
One twin strainer for lubricating oil of the engines, and
One twin strainer for the oil of the hydraulic coupling.

The advantages of this oil engine cruising plant can be easily visualized. It will permit to keep the steam plant idle whenever desired on long cruises, which will give a highly welcome relief to the operating personnel especially in tropics, but the main thing is that it will permit to cover long distances without refueling. The normal Diesel oil supply provided on these vessels will permit a cruising radius of 8000 miles, but if all available tanks be filled with Diesel oil, which naturally can also be burned under boilers, although under ordinary conditions it would be too expensive a boiler fuel, then the cruising radius reaches the most remarkable figure of 18,000 miles.

The necessarily considerable ship's requirements in electrical energy are provided by two turbine driven generators of 250 Kw. each, running at 3000 R.P.M., and two Diesel driven generator units of about 90 Kw. each operating at 1300 R.P.M. These four units are installed in three different locations so that there is no danger of failure of the entire electric plant should flooding occur. The generators are of direct current type, 220 volts pressure and of very light construction—7 kilograms (15.5 pounds) per B.H.P. The Diesel engines driving them are of the six-cylinder single-acting four-cycle type with air injection of fuel, built by Maybach Co., extensively for railroad service. The working cylinders are of 140 millimeter (5.512 inches) bore and 180 millimeters (7.087 inches) stroke and the engines are rated at 137.5 B.H.P. normal and 150 B.H.P. maximum.

To be able to furnish steam necessary for heating and bathing as well as for cooking either in port or when running on the cruising oil engines, without maintaining steam in the large propelling boilers, two small oil fired auxiliary boilers of 50 square meters (538 square feet) each are provided. One main and one auxiliary feed pump, a fuel oil pump, a feed water heater, a fuel oil tank and hot well complete this equipment.

A series of trials were made on cruising Diesel engines in addition to the standard trials, which are fully reported in the original article. The Diesel engines were capable of giving the ship a speed of 10.0 to 10.5 knots with the shafts turning 94 to 97 R.P.M. Only the total fuel consumption could be determined on these trials, but during the very extensive shop trials the Diesel engines demonstrated a fuel consumption of about 180–185 grams or .397–.408 pounds per B.H.P. per hour during a continuous run of many days. This result is without a doubt very acceptable for a new type of engine of such light construction and high speed. Based upon this figure the cruising radius of these cruisers becomes 8000 miles when Diesel bunkers only are used in 18,000 miles when all the boiler oil bunkers are also filled with Diesel oil. This compares favorably with the 5500 mile cruising radius of these vessels when running on steam turbines, although the cruising speed on steam is higher—15 knots.

OIL FROM COAL.

STATEMENT BY CHAIRMAN OF IMPERIAL CHEMICAL INDUSTRIES, LTD.

So much interest is being shown at the moment in the question of hydrogenating British coal into oil fuels, that, as chairman of the company which has done all the large-scale work on hydrogenation in this country, I may perhaps be allowed to clarify the position by making the following points.

Hydrogenation has already reached the stage at which we have a plant at Billingham capable of hydrogenating 15 tons or more per day of bituminous coal. Of the clean dry coal so treated some 60 per cent by weight is recovered as refined petrol, which has been utilized with complete success for many months past in motor-vehicles of all descriptions.

This fact, together with the experience which our associates in the recently formed International Hydrogenation Patents Company have already had in the production of 100,000 tons per year of petrol from brown coal and brown coal-tar, which experience is now fully available to us, enables me to state that: (1) Although high pressures and temperatures are used, our experience in the very similar technique of ammonia manufacture has enabled us to overcome satisfactorily all difficulties which have hitherto arisen. (2) The manufacture of hydrogen is a subject on which we have unique and extensive knowledge; neither the quantity required nor the price at which it can be produced presents any obstacle to the successful development of the hydrogenation process on an economic scale. (3) I am quite satisfied that it is now practicable to build in this country a plant for the production of petrol from bituminous coal. A plant to hydrogenate 1000 tons of clean dry coal per day could produce No. 1 motor spirit, and sell it at present standard retail prices with a small profit, after allowing for all costs of raw material, manufacture, repairs, obsolescence, and distribution.

It is not contended, however, that such a profit as the above would attract capital. It is mentioned as an indication of the fact that the hydrogenation process has already reached a stage at which only a very slight rise in the price of petrol is required to render it a self-supporting commercial proposition.—"Mechanical World and Engineering Record," Aug. 21, 1931.

THE EFFECT OF MODERN MACHINERY ON THE DESIGN OF LARGE SHIPS.

By A. T. WALL, O.B.E., and H. C. CAREY.

The speeds of transatlantic passenger-carrying vessels are always of importance and interest, and the present paper gives the results obtained by an investigation into the various possibilities.

The particular object of the paper is to show the overall advantage in the complete ship of improvements in modern machinery. The improvements which affect ship design to the greatest extent are (1) reduced weight of machinery, and (2) reduced fuel consumption. These are the factors which are dealt with primarily in this paper. Reduced space per horsepower also has obvious advantages, but although of great importance in the design of a particular ship, these advantages cannot be put into figures to cover all cases, but they are referred to in general terms and as necessary. The paper also shows the relation between speed and power, in so far as it is affected by length of ship, displacement coefficient, and weight of hull.

It is axiomatic that increase of weight, whether in hull, machinery, fuel, water, or other items, means loss of speed or increase of horsepower, other things being equal. Reduction of fuel costs on account of reduced consumption can be calculated directly in any given ship, and other economies inci-

dental to the smaller amount of fuel carried can also be assessed without much difficulty. The effect of reduced weight of machinery and fuel, however, cannot be easily assessed, but is more far-reaching, since it enables a smaller, cheaper, and more economical vessel to be designed, or, alternatively, it gives greater earning capacity for the same expenditure. Between these two limits are endless possible variations, but it is not necessary to consider all of these. They can be deduced from the results given.

Type of Ship.—The type of ship considered is representative of the modern transatlantic liner, on service between Southampton and New York, via Cherbourg. All vessels are of cruiser stern type, of normal form, with the usual arrangement of superstructure decks. In estimating weights, no attempt was made to depart from usual practice with regard to hull and outfit. There is certainly a possibility of weight economy in these items, but this is outside the scope of the paper, although the advantages of it are shown.

Machinery Types.—Three types of machinery have been selected, the principal variations being in the type of boiler and steam pressure and temperatures. Diesel machinery has not been considered, since the state of its development does not yet warrant consideration for high powers. In all cases the engines are single reduction three-stage Parsons turbines on each shaft, with two intermediate pressure turbines for powers of 30,000 shaft horsepower per shaft and over. The closed-feed system, with regenerative condensers, is used for all types, and a vacuum of 29 inches is assumed throughout. For the smaller powers a twin-screw arrangement is taken, and quadruple screws for the larger powers. The boilers are of three types:—(*a*) Double-ended cylindrical Scotch, with triple-flow air heaters and forced draught. (*b*) Yarrow, five drum, double-flow, side-fired, with auxiliary boilers for feed make-up on the Johnson system, quadruple-flow air heaters and forced draught. (*c*) Johnson two drum, with auxiliary cylindrical boilers as in (*b*), quadruple-flow air heaters, and forced and induced draught. Particulars of pressures, temperature, and efficiencies of the three types are given in Table I.

TABLE I.

Type of Boiler.	Type A. Cylindrical.	Type B. Yarrow.	Type C. Johnson.
Steam pressure, lb. per sq. in.	250	450	450
Superheat in deg. F.	244	290	290
Final steam temperature, deg. F.	650	750	750
Boiler efficiency (per cent.)	80	85	87
Feed temperature, deg. F.	200	300	300

Other types of boilers, such as the Babcock and Wilcox ordinary and S.X. types, might also have been considered, but sufficient ground is covered by the three types mentioned. The auxiliary machinery has been varied slightly between the three types, and a description is given in Appendix I, together with typical machinery weights and fuel consumption.

Type A represents good modern practice with cylindrical boilers, with moderate pressure and temperatures, and forms the basis of comparison. Type B represents modern general practice for high-speed ships, except that the boiler ratings have been taken in excess of the usual 5 to 5½ pounds now adoptd for marine work. They vary from 6.8 pounds of steam (actual)

per square foot of heating surface for a boiler with output of 66,000 pounds of steam per hour to 7.2 pounds in a boiler with an output of 110,000 pounds per hour. The makers recommend these ratings as being well within practical limits without jeopardizing reliability. Type C might be said to represent ultra-modern practice, in that the Johnson boiler is the latest water-tube boiler to be applied successfully to marine propulsion. The ratings in this case are taken as varying from 11.4 to 12 pounds per square foot of heating surface for boiler outputs of 66,000 and 110,000 pounds of steam per hour, respectively. These ratings are taken from information supplied by the licensees, Messrs. Clarke, Chapman and Company, Limited.

Basis of Calculations.—To obtain all the necessary data, ships of various sizes, displacement coefficients, and speeds were considered. Four lengths were taken, each length being used in conjunction with four displacement coefficients of 0.47, 0.54, 0.61 and 0.68, based on length between perpendiculars, and speeds of 20 knots and upwards. Average proportions for beam, depth, and draught were derived from average present practice, and the dimensions (in feet) of the vessels considered are as shown in Table II. For

TABLE II.

	Ft.	Ft.	Ft.	Ft.
Length (between perpendiculars)	550	700	850	1,000
Length (on water-line) ..	572	728	884	1,040
Breadth (moulded) ..	74	86	98	110
Depth (to uppermost continuous deck) ..	50	56	62	68
Load draught	29	32	35	38

each combination of length, displacement coefficient, and speed, the horse-powers were obtained by the use of Taylor's Standard Series. For a fuller description of the method used reference should be made to a paper entitled "Some Considerations in the Design of Channel Steamers," read before the Liverpool Engineering Society in 1915, by Mr. A. T. Wall.

Effect of Length on Speed.—The most economical ship for the deadweight and speed in Table IV would lie between the limits shown in (*a*) and (*b*),

TABLE IV.—*Vessels Carrying* 2,000 *Tons Deadweight at* 29 *Knots.*

——	Length ft.).	Displacement Coefficient.	Displacement (tons).	S.H.P.
	(*a*) *Minimum Length.*			
Cylindrical boilers	923	0·642	67,140	189,500
Yarrow boilers..	738	0·595	36,600	128,500
Johnson boilers	692	0·577	31,010	111,500
	(*b*) *Minimum Shaft Horse-power.*			
Cylindrical boilers	1,011	0·538	65,860	139,000
Yarrow boilers..	804	0·522	38,560	101,000
Johnson boilers	762	0·511	33,650	92,000.

for each type of machinery, because as the length in (*b*) is decreased, the hull cost decreases, while the machinery and fuel costs increase. There will be some intermediate length which gives the best combination of first cost and running expenses for the maximum economy. The most economical length will generally be nearer the length for minimum horsepower than the minimum length, since reduction of shaft horsepower and fuel consumption represents a large value when capitalized over the life of the ship. It will be noted with cylindrical boilers that the minimum length of 923 feet, compared with the length of 1011 feet for minimum power, is only made possible by an increase of displacement of 2 per cent, even although the ship is shorter, and the shaft horsepower is increased by over 36 per cent. The effect with Yarrow and Johnson boilers is not quite so marked, the displacements with the smaller lengths being actually decreased by 5 per cent and nearly 8 per cent, respectively, while the shaft horsepowers are increased by about 27 per cent and 21 per cent, respectively. In other words, the increase of shaft horsepower and consequent fuel consumption with reduced length is not so great with the more modern machinery, so that minimum hull dimensions, and therefore initial costs, more nearly approach the length for minimum running costs—a distinct advantage.

Even taking the direct mean of minimum length and length for minimum horsepower, the ship with cylindrical boilers would be 967 feet long as compared with 727 feet for the ship with Johnson boilers. The corresponding shaft horsepowers, taken from curves, are 142,500 and 93,500 shaft horsepower on displacement coefficients of 0.564 and 0.529, respectively. From these figures, the adoption of Johnson instead of cylindrical boilers shows a saving of 24.8 per cent in length, about 50 per cent in hull cost, and 34.4 per cent in horsepower and fuel consumption. These figures apply to the case taken, viz., a speed of 29 knots and carrying 2000 tons deadweight, but at other speeds and deadweight also large economies would be effected.

General Results.—The results of these investigations are summarized in the following paragraphs. When the word "ship" is used, it refers to ships of the type and for the trade described in the paper. (1) With length limited to 1000 feet, the maximum speed that could be reached, if the necessary power could be transmitted through the machinery, is about 36½ knots. With present-day limits of transmission of power, the maximum speed with quadruple screws and 240,000 total shaft horsepower is about 35 knots. These speeds can only be obtained by the use of Johnson boilers, giving a machinery weight corresponding to about 22 shaft horsepower per ton. The corresponding speeds with Yarrow boilers are about 35 and 34 knots, respectively. (2) The maximum horsepower likely to be installed in larger liners will not exceed 240,000 shaft horsepower for some time to come on account of transmission limitations. (3) Modern steam producers are so efficient that space can be found in any ship for the necessary boilers to drive her at her maximum possible speed. (4) Cylindrical boilers will limit the maximum possible speeds for ships of lengths greater than about 855 feet, since the available space is insufficient to house the boilers, although the necessary weight could be carried on these lengths. (5) Generally higher speeds are obtainable and maximum possible speeds are greater with modern steam producers than with the old, varying from 3.3 knots increase in a 550-foot ship to 5.6 knots in a 1000-foot ship, in changing from cylindrical to Johnson boilers.

(6) It is most uneconomical to drive a ship at her maximum possible speed with any type of machinery. (7) For a given speed and deadweight there is a minimum length of ship coupled with a certain power, and a minimum horsepower coupled with a greater length. The most economical ship will have a length and horsepower between these two values, respectively. These values can be found by the methods given. (8) Higher speeds

must involve longer and larger ships for the greatest overall economy, and the initial and running costs increase quite out of proportion to the percentage increase of speed. (9) Appreciable economy is obtained by reducing weight, whether in hull, machinery, or fuel per shaft horsepower. (10) Modern machinery permits the adoption of smaller displacement coefficients than with the old, giving improvement in performance at sea.

(11) Modern machinery enables size and horsepower for a given deadweight and speed to be reduced. A particular case is quoted above resulting in reductions of 24.8 per cent in length, 50 per cent in hull cost, and 34.4 per cent in horsepower and fuel consumption. (12) With modern machinery minimum hull dimensions for given conditions more nearly approach those required for minimum horsepower, so that the loss by the necessary compromise is reduced. (13) Modern machinery will affect hull design, since to carry the same number of passengers as a larger ship with old machinery, superstructures will be increased. Passenger accommodation will therefore generally be higher in the ship, and consequently better. (14) Modern machinery enables greater speeds to be obtained on the same draught; a definite advantage when related to the limitation of depth of water in some harbors which cannot be deepened without considerable expense. (15) Generally, great economies are possible with the use of modern machinery both in first cost of hull and machinery and in running.

TABLE V.

Reference Number.	1.	2.	3.	4.
Length of ship (ft.) ..	550	700	850	1,000
Speed (knots)	20	25	30	32
Maximum S.H.P. ..	28,000	67,500	183,000	260,500
Number of shafts ..	2	2	4	4
Propeller revolutions per minute	160	180	245	240

TABLE VI.

Machinery Type.	Reference No.	Machinery Oil Fuel. Weight (tons).	Machinery Oil Fuel. Consumption, lb. per S.H.P. per hour.	S.H.P. per Ton Weight.
A. (Cylindrical boilers) ..	1	3,402	0·696	8·23
	2	7,010	0·655	9·63
	3	16,760	0·622	10·92
	4	22,910	0·608	11·36
B. (Yarrow boilers) ..	1	2,345	0·604	11·94
	2	4,580	0·569	14·75
	3	10,000	0·544	18·30
	4	13,302	0·536	19·58
C. (Johnson boilers) ..	1	2,133	0·593	13·12
	2	4,120	0·558	16·40
	3	8,950	0·534	2 45
	4	11,890	0 526	21 91

APPENDIX I.

Main Machinery.—Main turbines are as described in the paper. For powers up to 30,000 shaft horsepower per shaft, the astern turbines consist of an impulse wheel in the intermediate pressure casing, in series with a low-pressure astern turbine of the impulse reaction type in the exhaust end of the low-pressure ahead casing. For powers of 30,000 shaft horsepower and over, the ahead low-pressure turbine is of double-flow design, and the high-pressure and low-pressure astern turbines are incorporated in the second intermediate-pressure and low-pressure casings, respectively. Turbine and propeller revolutions have been worked out for each particular case, the range of propeller revolutions being from 130 R.P.M. to 300 R.P.M. Shafting arrangements and weights follow normal practice.

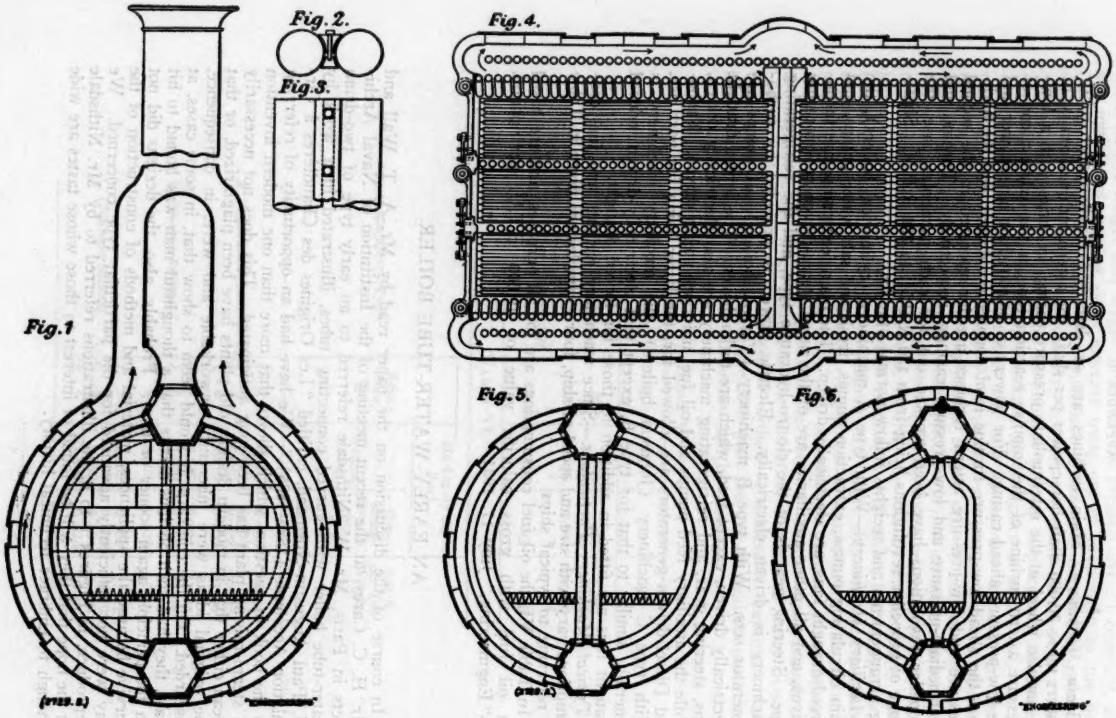
Auxiliary Machinery.—With type A machinery (cylindrical boilers), the main circulating pumps, extraction pumps, fresh and salt water pumps, and forced draught fans, are electrically driven. The main feed pumps are turbo-driven, and all other auxiliaries are of the usual steam-driven reciprocating type. Steering gear is of the electro-hydraulic type, and the refrigerating machinery is driven electrically. Electric power is provided by turbo-generator sets. With type B machinery (Yarrow boilers), all pumps are electrically driven, except feed, which are turbo-driven. The forced draught fans, steering gear, and refrigerating machinery are also electrically driven, while the auxiliary feed pumps and oil fuel units are steam driven. Turbo- and Diesel-driven generators are allowed for, the latter being for port use. With type C machinery (Johnson boilers), the auxiliary machinery is generally similar to that for type B, except that electrically-driven induced draught fans are fitted in addition to those for forced draught.

Typical Machinery Particulars.—Since machinery weights and fuel consumptions vary with size and speed of ship, four examples are given in Table V referring to typical ships.

In Table VI, the oil fuel consumptions are for all purposes, and are based on oil fuel with gross calorific value of 19,000 B.T.U. per pound. —"Engineering," July 17, 1931.

AN EARLY WATER-TUBE BOILER.

In course of the discussion on the paper read by Mr. A. T. Wall and Mr. H. C. Carey, at the recent meeting of the Institution of Naval Architects in Paris, Mr. W. Nithsdale referred to an early type of two-drum water-tube boiler with curved connecting tubes, illustrated in a work by M. Paul Augustin-Normand, entitled "Les Origines des Chaudières à Circulation Accélérée (1825-1885)." We have had an opportunity of referring to the book in question, which shows that more than one modern invention in steam-raising plant has been long anticipated. This does not necessarily mean either that in modern designs old plants have been plagiarized, or that the original ideas were inherently unworkable and were, in consequence, discarded. On the contrary, it would seem to show that, in some cases, at least, they had such obvious merits that a thoughtful man was bound to hit on the principle again sooner or later. Probably, also, the designs did not survive because the appliances, material and methods of construction of the day were not sufficiently advanced for the particular type concerned. We reproduce, in Figures 1 to 6, the illustrations referred to by Mr. Nithsdale in the hope that they will be found of interest to those whose tastes are wide enough to embrace technical history.



CRADDOCK WATERTUBE BOILER OF 1857.

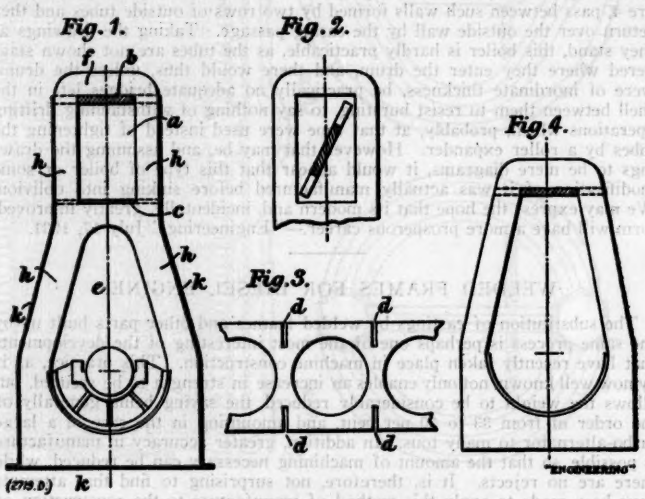
Two types of boiler are shown, both of which are assigned to Thomas Craddock and are dated 1857. Figure 1 shows a cross-section of a boiler which has, apparently, a dry back. The tubes, as shown in Figures 2 and 3, seem to have the intervening spaces filled in, so as to form a solid wall, the gases finding their way back past the tube walls surrounding the furnace by way of an annular space between the walls and the enclosing casing, and thence escaping to the chimney. Figures 4, 5 and 6 show sections of a double-ended boiler, with a central vertical nest of tubes. In one section this nest consists of two rows of tubes close together, in the other the nest is divided and contains a grate. The tubes are very closely pitched so as to form practically continuous walls, and the gases, as shown in the sectional plan in Figure 4, pass between such walls formed by two rows of outside tubes and then return over the outside wall by the casing passage. Taking the drawings as they stand, this boiler is hardly practicable, as the tubes are not shown staggered where they enter the drum, and there would thus, unless the drums were of inordinate thickness, be practically no adequate bridges left in the shell between them to resist bursting, to say nothing of withstanding drifting operations which, probably, at that time were used instead of tightening the tubes by a roller expander. However that may be, and assuming the drawings to be mere diagrams, it would appear that this type of boiler or some modification of it was actually manufactured before sinking into oblivion. We may express the hope that its modern and, incidentally, greatly improved, form will have a more prosperous career.—"Engineering," July 17, 1931.

WELDED FRAMES FOR DIESEL ENGINES.

The substitution of castings by welded frames and other parts built up by the same process is perhaps one of the most interesting of the developments that have recently taken place in machine construction. This practice, as is by now well known, not only enables an increase in strength to be secured, but allows the weight to be considerably reduced, the saving being generally of the order of from 33 to 50 per cent, and amounting in the case of a large turbo-alternator to many tons. In addition, greater accuracy in manufacture is possible, so that the amount of machining necessary can be reduced, while there are no rejects. It is, therefore, not surprising to find that attempts have been made to apply this method of manufacture to the construction of Diesel engines, the great weight of which when referred to their output is often a distinct disadvantage. This particularly applies, of course, in marine work and when they are used for transport purposes, as in these cases the weight may be a deciding factor in determining whether they shall be employed in preference to other prime movers. This being the position, we may therefore draw attention to the system of construction which has been patented by Mr. C. H. Stevens, of the Steel Barrel Company, Limited, Uxbridge, and has been adopted by Messrs. Davey, Paxman, Limited, Colchester, in their latest design of marine Diesel engine, a step which should prove of great interest to naval and other marine engineers. We understand, in fact, that this development is being closely watched by the British Admiralty, who have already one set of welded steel columns, designed by Mr. Stevens, for a new submarine engine, under construction at the Steel Barrel Company's works.

The underlying feature of the design is that all the explosion stresses are taken up by a series of main frame plates, one of which is located between each pair of cylinders. These are so arranged that they act as a sling round the cylinder head seatings and crankshaft. It is, however, distinguished from the somewhat similar arrangement in which the upper end of a plate frame of this kind is secured to an entablature on which the cylinders

are supported, since no horizontal joints are used and the trouble which arises owing to the tendency of the bolts to fail under the severe tensile stresses that are set up at the time of the explosions is therefore eliminated. An elevation of one of these frame plates is shown in diagrammatic form in Figure 1. This is cut from a single rolled steel plate *h*, so that its contour in elevation corresponds with that of the end view of the engine. An opening *a* is next cut in the top part of this plate through which two plates *b* and *c*, running the whole length of the engine, are inserted. The upper of these two plates forms the seating for the cylinder heads, while the other provides a housing for the lower end of the cylinder liners. The way in which



WELDED DIESEL-ENGINE FRAMES.

MESSRS. THE STEEL BARREL COMPANY, LIMITED, ENGINEERS, UXBRIDGE.

these plates are inserted is shown in Figure 2, and they are slotted, as indicated in Figure 3, at *d*, to accommodate the side and tension members of the main frame plates. A second opening *e* in the lower part of the frame plate is provided for the insertion of the crankshaft bearing which may, of course, be either integral with the frame or a separate unit. The series of frame plates built up in this way are connected by a wrapper plate *k*, which extends the whole length of the engine on both sides and which may be provided with such doors and hand holes as may be required. The result is that the portion of the frame plate above the plate *b* acts as a girder, which receives the upward thrust due to the explosions in the cylinders, while the base forms a second girder, which similarly takes up the downward thrust. It will also be noted that all the usual horizontal bolted joints are omitted and that the use of through bolts as tension members is avoided. It may be added that when the cylinders and water jackets are cast in one piece the plate *c* can be dispensed with, the frame then being built up in the way shown in Figure 4.

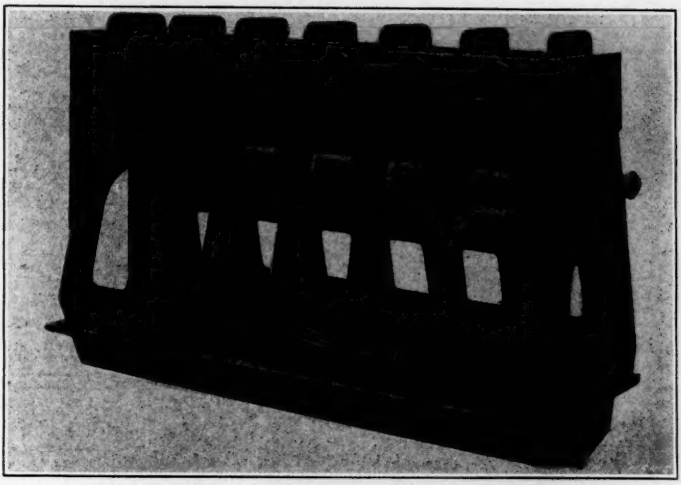


FIGURE 5.

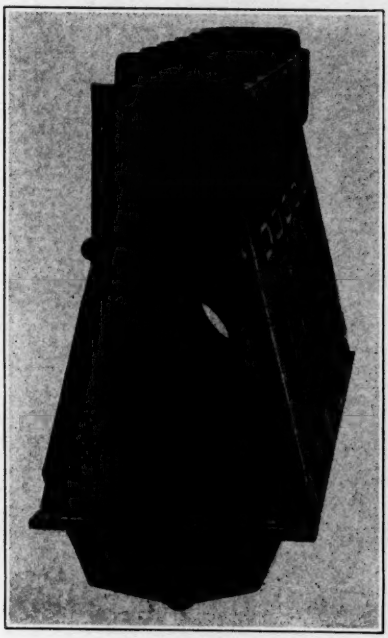


FIGURE 7.

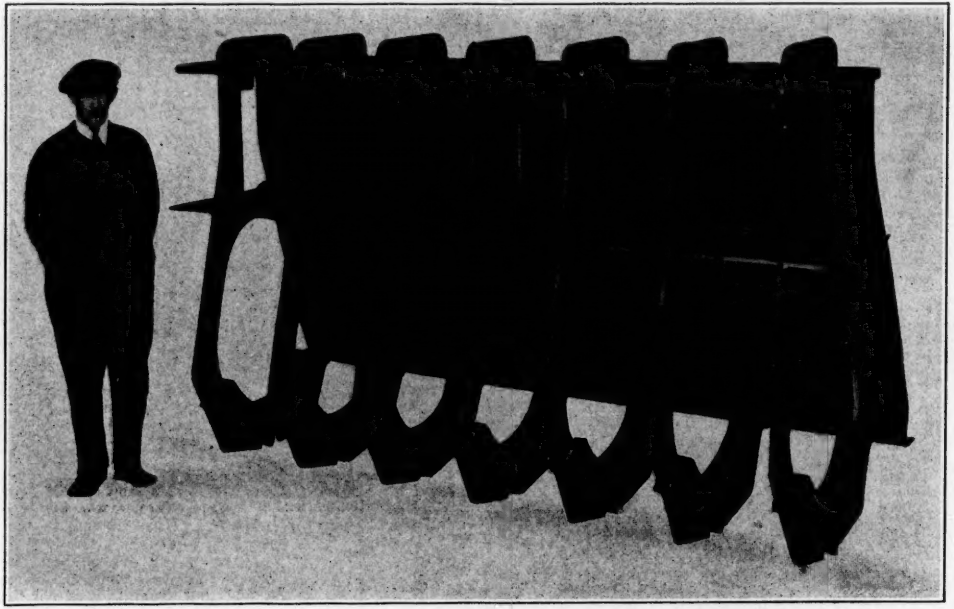


FIGURE 6.

The various parts of the frame which have just been described are welded together electrically as they are assembled, an important feature of the arrangement being that none of the welds which join the longitudinal cylinder plates to the vertical frame plates are under stress, since all the primary stresses are taken up by the top and bottom girders and the side tension members of the main frame plates.

The application of this method of construction to a six-cylinder Diesel engine will be clear from Figure 6, which shows all the vertical frame plates in position, the upper and lower cylinder plates welded in and one wrapper plate welded on. The bosses for the cylinder head studs, the lower liner housing and the seatings for the main bearings are also visible. Side and end views of the same engine are given in Figures 6 and 7, respectively. This engine frame, which has been designed by Messrs. Davey, Paxman, Limited, Colchester, under license from Mr. Stevens, and manufactured by the Steel Barrel Company, Limited, Uxbridge, makes the total engine weight 27 pounds per horsepower compared with the 48 pounds per horsepower which would have been reached had castings been used. The former figure can, doubtless, be still further reduced when more experience has been gained.—"Engineering," July 17, 1931.

ARMISTICE DAY.

The following anecdote contributed by Lieut. G. M. Dusinberre, U. S. N., while not quite in the spirit of the JOURNAL, may be considered worth preserving in its pages on account of the distinctly engineering flavor. It has the merit of being true.

In 1921 there was an Alnav to the effect that Armistice Day would be observed by ceasing all naval activities at noon and holding two minutes of silent prayer.

The destroyer *Childs* was then underway from Cherbourg to Gibraltar, and the Captain decided that the ship should be stopped during the specified period.

On inspecting the rough log sheet that afternoon, the Engineer Officer observed the following entry:

"12 to 4 p. m.

"Steaming as before, standard speed 15 knots, 165 r.p.m., main engines and Nos. 1 & 2 boilers in use, auxiliary exhaust to H.P. turbine.

"At 12:00 stopped main engines, and shifted auxiliary exhaust to main condenser, in commemoration of World War dead."

BOOK REVIEWS.

STEAM TURBINE OPERATION. By William J. Kearton, M. Eng. Isaac Pitman & Sons, Publishers, 2 West 45th Street, New York. Price $3.75.

As the title indicates this is a book on Turbine Operation and its object is well expressed by the Author in the preface.

"The author has not restricted himself to setting out mere instructions respecting the various phases of the installation, running, and maintenance of steam turbines, but rather he has attempted to include such descriptive matter as will enable the operating engineer better to understand the construction and functioning of the plant under his care, and also to present an account of those thermal and mechanical considerations which affect the operation and handling of larger machines."

That this object has been attained is evident from a study of the book which consists of ten chapters, 293 pages of text and many excellent illustrations.

A valuable book for any engineer.

O. L. Cox.

SCHIFFBAU KALENDER—1931. By W. Gűtschow and Hermann Hildebrandt. 474 Pages, 196 Figures, Flexible Linen Binding. Price RM-12.00 ($3.00). Strauss, Vetter & Co., Berlin SW68, Neuenburger Str. 8.

This compact and handy little book is not a calendar at all; it is an excellent collection of much pertinent and up-to-date information on ships. The title, which may appear misleading to an American engineer, is the only remaining heritage of many predecessors of this book, which was however, thoroughly rewritten by its present distinguished authors.

The first part of the book presents much information of general nature such as mathematical tables, weights and measures, fol-

lowed by chapters on materials and machine details, theoretical mathematics and mechanics, ship theory, strength of materials. A chapter dealing with rules and regulations of ship operation is followed by one on ship design, which also gives brief but very complete descriptions of many examples of most modern types of ships such as express liners, fast freighters, ore-ships, coastwise vessels and many different types of motorships both of German and foreign design. Resistance of ships is dealt with in one chapter, while another chapter is devoted to interior arrangement of ships and includes details of life boats, rudders, steering apparatus, etc. The chapter on propelling machinery deals briefly with existing types of drives and types of propelling engines. The last chapter covers economies of ship operation.

The book deserves the attention of all people concerned with the design building and operation of ships. It is written in German.

E. C. MAGDEBURGER.

ASSOCIATION NOTES.

ANNUAL BUSINESS MEETING.

The Annual business meeting of the Society was held at Washington, D. C., on Tuesday, October 6, 1931. The following were nominated for offices for the year 1932:

For President:

Rear Admiral Frank B. Upham, U. S. N.

For Secretary-Treasurer:

Captain O. L. Cox, U. S. N.
Commander H. F. D. Davis, U. S. N.

For Member of the Council:

Rear Admiral S. M. Robinson, U. S. N.
Rear Admiral George C. Day, U. S. N.
Captain H. G. Bowen, U. S. N.
Captain R. C. Davis, U. S. N.
Captain E. G. Oberlin, U. S. N.
Lieut. Commander T. A. Solberg, U. S. N.
Rear Admiral W. L. Capps (C. C.), U. S. N.
Captain H. S. Howard (C. C.), U. S. N.
Captain J. Q. Walton, U. S. C. G.
Mr. B. P. Lamberton.
Mr. C. P. Wetherbee.
Mr. W. M. McFarland.
Mr. J. F. Metten.

Ballots are in the hands of voting membership and polls will close at 4:30 P. M. on 26 December, 1931.

ANNUAL BANQUET.

It has been decided to hold another banquet of the Society early in 1932. Details will be promulgated in the near future.

MEMBERSHIP.

The following have joined the Society since the publication of the last previous JOURNAL:

NAVAL.

Aubrey, Norbert, Lieutenant, U. S. N. R.
Dannarth, Carl, Ensign, U. S. N. R.
Hansen, E. H., Lieutenant, U. S. N.
Jordan, Beckwith, Lieutenant, U. S. C. G.
Upham, Frank B., Rear Admiral, U. S. N.

CIVIL.

Amos, J. K., Engineering Experiment Station, Annapolis, Md.

Anoshenko, B. V., Westinghouse Electric and Manufacturing Co., Lester P. O., Philadelphia, Pa.

Buddeman, Carl E., Suite 1531, 111 West Washington St., Chicago, Ill.

Karelitz, G. B., Westinghouse Electric and Manufacturing Co., Lester P. O., Philadelphia, Pa.

Davis, R. H., 1616 Walnut St., Philadelphia, Pa.

Lanteri, Albert, 357 87th St., Brooklyn, N. Y.

Largergren, G. M., Pusey and Jones Co., Wilmington, Del.

Lee, C. Stuart, Pusey and Jones Co., Wilmington, Del.

Marsland, R., Westinghouse Electric and Manufacturing Co., Lester P. O., Philadelphia, Pa.

Ronay, Bela N., Engineering Experiment Station, Annapolis, Md.

Spiegelhalter, A. G., Pusey and Jones Co., Wilmington, Del.

Truss, William M., Pusey and Jones Co., Wilmington, Del.

ASSOCIATE.

Manuel, Hemogenes S., Box 426, University Station, Tucson, Arizona.